[redacted]

WILEY ENGINEERING SERIES—No. 1

POWER TRANSMISSION BY LEATHER BELTING

By

ROBERT THURSTON KENT, M.E.

CONSULTING ENGINEER

Junior, American Society of Mechanical Engineers

FIRST EDITION

NEW YORK

JOHN WILEY & SONS, INC.

LONDON: CHAPMAN & HALL, LIMITED

1916

PRESS OF
BRAUNWORTH & CO.
BOOK MANUFACTURERS
BROOKLYN, N. Y.

Engin.- Replace
Wah-r
10-23-25
12420

PREFACE

THE advent of high-speed steel and of intensive methods of production has rendered the problem of belt maintenance one of the most important of the many that the factory manager has to solve. In the machine shop belts must be proportioned to pull the heavier loads that are used in modern practice, and in any industry the belts must be so taken care of that the interruptions to manufacture due to belt failures will be reduced to the minimum. Interruptions to manufacture mean loss of production and loss of profits.

Concurrently with the development of improved methods of production there has grown up an improved system of belting practice, which has kept pace with production. The literature of these improved belting methods is buried in the transactions of engineering societies and in the files of technical journals. It is so scattered that it is difficult for the average man to comprehend that the art of power transmission by means of leather belting has completely changed in the past ten or fifteen years. The object of this book has been to gather together the best information on the new practice and compile it in a form that would be of the greatest service to the belting user.

With the exception of the calculation of the tables that form a part of the work, the author makes no claim to originality. His office has simply been that of compiler of the work of Taylor, Barth and others. If the work that has been done in preparing this book will lead to better belting practice in the shops of the country, it will have accomplished its object.

ROBERT THURSTON KENT.

MONTCLAIR, N. J.,
March 9, 1916.

WILEY ENGINEERING SERIES

The Wiley Engineering Series will embrace books devoted to single subjects. The object of the Series is to place in the hands of the reader all the essential information regarding the particular subject in which he may be interested. Extraneous topics are excluded, and the contents of each book are confined to the field indicated by its title.

It has been considered advisable to make these books manuals of practice, rather than theoretical discussions of the subjects treated. The theory is fully discussed in text-books, hence the engineer who has previously mastered it there, is, as a rule, more interested in the practice. The Wiley Engineering Series therefore will present the most approved practice, with only such theoretical discussion as may be necessary to elucidate such practice.

TABLE OF CONTENTS

HOW TO USE THIS BOOK

THE shop engineer, the master mechanic, and the man in charge of the belts in the shop will find Chapters III, V, and VI and the tables in the back of the book of the greatest service. These contain all the necessary information for the selection of the proper size of belts to transmit a given horsepower, the tension at which they should be operated, the length of time that should elapse between retightenings, and notes on the proper care and maintenance of belts.

For the proper arrangement of pulleys and the laying out of belt drives the notes in Chapter VIII will be of service.

The theory on which the practice given in the foregoing chapters is based will be found in Chapters I and IV. This is of interest to the student and the teacher, but is not necessary for the practical man.

The history of the experiments on which both theory and practice are based is presented in Chapter II.

The shop man will also find Chapter VII of value, as it contains information regarding the quality of belting, methods of utilizing old belting, recipes for belt dressings, etc., belting mathematics of use in the shop, and other practical information.

POWER TRANSMISSION BY LEATHER BELTING

CHAPTER I

GENERAL CONSIDERATIONS AFFECTING THE TRANSMISSION OF POWER BY LEATHER BELTING

The function of a belt drive is to transmit a certain amount of power from one pulley, rotating at a given number of revolutions per minute, to another pulley rotating at the same or a different speed. The amount of power transmitted will depend upon the velocity of the belt, the tension under which it is placed on the pulleys, the arc of the smaller pulley with which the belt is in contact and the conditions under which the belt is used. As the tension in the belt in service decreases, the amount of power that will be transmitted by it will decrease, and therefore an important consideration in laying out a belt drive is the determination of the minimum tension allowable in a belt which is to do a predetermined amount of work.

That belt drive is the most satisfactory which in the long run will transmit the desired power at the lowest total cost. Included in the total cost of the belt are: The first cost; the cost of repairs and maintenance; and the loss caused by the loss of production due to machinery shutdowns as a result of belt failures during working hours. The importance of this latter item has not been appreciated to the extent that it deserves, despite the fact that it often is the most important of the three. A little consideration will make

clear that a machine whose earning capacity is high will lose for its owners a considerable sum of money if its production is stopped during working hours. This loss of production, coupled with the wages of the operator of the machine during the period that it is idle, will often far exceed the cost of the heavier belt and the expense of regular inspection and maintenance that would have avoided the belt failure. To obtain the lowest possible total belting cost, therefore, not only is good design necessary, but also provision for regular inspection of all the belts in a shop and their maintenance in good order, together with regulation of the tensions under which they are required to do their work.

Relation of Sizes and Speeds of Pulleys. In laying out a belt drive, the diameter and speed of rotation of one of the pulleys involved is usually a fixed quantity, and also the speed of rotation of the other pulley. Thus, if a machine is to be driven, the number of revolutions at which it is designed to operate is as a rule fixed by the maker, and the driving pulley cannot be varied in diameter. The speed of the line or countershaft from which it is driven is also incapable of variation. Or, in the case of a motor or engine driving a line shaft the pulley or fly wheel on the motor or engine is fixed in size, as is the number of revolutions per minute at which these prime movers operate. The speed of the line shaft is also fixed by shop conditions. The first step, therefore, in laying out the drive is the determination of the size of the pulley whose diameter is unknown, so that the shaft or machine will be driven at the desired speed.

The speed of rotation of two shafts connected by a belt varies directly as the diameters of the pulleys. If D and d are the diameters of two pulleys, both in the same unit, as feet or inches, and R and r their respective speeds of rotation in revolutions per minute, then

$$DR = dr.$$

If D, R and r are given

$$d=\frac{DR}{r}.$$

Example: A motor running at 1000 r.p.m. has a pulley of 6 in. diameter, and is required to drive a line shaft at a speed of 250 r.p.m. What size of pulley is required on the line shaft?

Substituting in the above formula, we have

$$d=\frac{1000\times 6}{250}=24 \text{ in.}$$

Size of Belt Required. The pulley sizes being known the question next to be answered is " How large a belt must be used to transmit the horsepower needed? " The width of belt that can be used is usually limited by the width of the pulley on the machine or motor, in the case of a primary or final drive, and only the thickness and tension in the belt can be varied to suit the conditions imposed. In the case of the drive from a line shaft to a jack shaft or to a countershaft the designer, as a rule, has no such limitations, and can adopt any combination of width, thickness and tension as best suits his convenience or the pulleys and belts at his disposal. A belt can, within the practical range of belt velocities, be made to transmit almost any amount of power desired, up to the limit of its strength, by increasing the tension under which it operates. High belt tensions, however, mean high bearing and journal pressures, resulting in an increased expenditure of power for overcoming line shaft friction; they also mean high maintenance and repair charges for the belts, frequent belt failures and short life of belts. Thus while the adoption of a high tension will permit the use of a thinner and lighter belt, the saving in first cost will be more than counterbalanced by the high maintenance and replacement charges and the generally unsatisfactory service that will result. The use of heavy belts and comparatively

low tensions is in accord with the best belt practice of today, and is recommended as giving in the end the lowest total belting cost as well as the most satisfactory service.

The horsepower that will be transmitted by a belt can be expressed by the general formula

$$\text{H.P.} = \frac{pVA}{33000},$$

in which p is the effective pull in the belt, or the difference in tension between the tight and slack strands of the running belt, in pounds per square inch; V is the velocity of the belt in feet per minute; and A the cross-sectional area of the belt in square inches. By assuming a value for p and for the thickness of the belt, the formula can be made to read

$$\text{H.P.} = \frac{wV}{C},$$

w being the width of the belt in inches and C a constant whose value depends on the thickness of the belt and the value adopted for p. This formula, whose use as it stands is not to be recommended, is the basis of the old rules of thumb for determining the size of belt required for a given horsepower. One such typical rule, the value of p being assumed as 360, reads " A single belt, 1 in. wide, traveling at the rate of 550 ft. per minute, will transmit one horsepower."

Faults of the Rules of Thumb. These rules of thumb are all defective in that they neglect many of the factors affecting the value of p. The effective pull depends on the arc of contact of the belt on the pulley, on the coefficient of friction between the belt and the surface of the pulley, on the initial tension of the belt when at rest, and on its velocity. As the velocity of the belt increases, the centrifugal force developed by the belt in passing over the pulleys manifests itself and tends to decrease the effective pull. This effect, known as centrifugal tension, is serious at high velocities and

limits the effective pull that can be developed. It causes the horsepower that can be transmitted, within the range of initial tensions that are recommended, to reach a maximum at a velocity of about 4000 ft. per minute. Any increase in velocity beyond this point actually decreases the horsepower that will be transmitted by the belt, and at a velocity in the neighborhood of 6800 ft. per minute the belt will fail to transmit any power whatever. The effect of centrifugal tension is entirely neglected in the rules of thumb; according to them the horsepower transmitted will increase indefinitely with an increase in velocity. Also, in the rules of thumb the coefficient of friction is assumed to be a constant quantity at all velocities. As a matter of fact, it is shown by the studies of the experiments of Wilfred Lewis by Carl G. Barth that the coefficient of friction is a variable quantity that increases with an increase in velocity.

It is thus evident that the rules of thumb are unreliable even for determining the horsepower that a belt will transmit, and that they give no information as to the tension under which belts should be put upon the pulleys. It is also evident that the design of a belt drive is not a simple matter of substitution in a formula, but requires the careful consideration of many factors.

Recent Investigations on Belts. Mr. A. F. Nagle[1] was one of the first to present a means of determining the horsepower that could be transmitted by a belt, which took into consideration the different variables enumerated above. Mr. Nagle, however, assumed a value for the coefficient of friction that was constant at all velocities of belt, and also based the value of the effective pull which could be allowed in the belt upon the strength of the joint. This procedure gave working stresses in the belt 275 lb. per square inch for laced belts and of 400 lb. per square inch for riveted belts.

[1] Transactions, American Society of Mechanical Engineers, vol. ii, p. 91.

While the use of Nagle's formula will give better results than will the rule of thumb formulæ, nevertheless the tensions in the belt will be much higher than are dictated by the good practice which has resulted from later investigations, notably those of Frederick W. Taylor, supplemented by the further studies of Carl G. Barth.

While the work of Mr. Nagle represented a long step in advance in belting practice, the investigations of Taylor,[1] extending over nine years, and dealing with belts in actual service, really form the basis upon which the most approved practice of today is built. These investigations revealed the advantages of low tensions and heavy belts, the accurate measurement of the tension under which the belt is placed on the pulleys, the regular inspection, repair and retightening of belts, and the maintenance of the working tension between certain well defined limits. Taylor's investigations are discussed in greater detail in Chapter II.

Taking Mr. Taylor's results as a starting point, and examining the work of other reliable investigators in connection with them, Carl G. Barth[2] has extended the work of Taylor, which applied mainly to slow-running belts, to high-speed belts such as are found in general service, and at the same time he has developed a mathematical treatment of the subject, which is the most accurate of any yet presented. As a result of the work of Barth, belting practice can now be said to be standardized, and the proportioning of a belt drive removed from the domain of rule of thumb to that of exact science. Mr. Barth's work is presented in detail in Chapter IV, and tables showing the horsepower that can be transmitted by a belt under different conditions of service, as calculated according to the formulæ derived by him, are presented on pages 91 to 94. The practice of Barth in

[1] Transactions, American Society of Mechanical Engineers, vol. xv, p. 204.
[2] *Ibid.*, vol. xxxi, p. 29.

proportioning belt drives is the practice recommended by the author.

Fig. 1 presents curves showing the horsepower transmitted

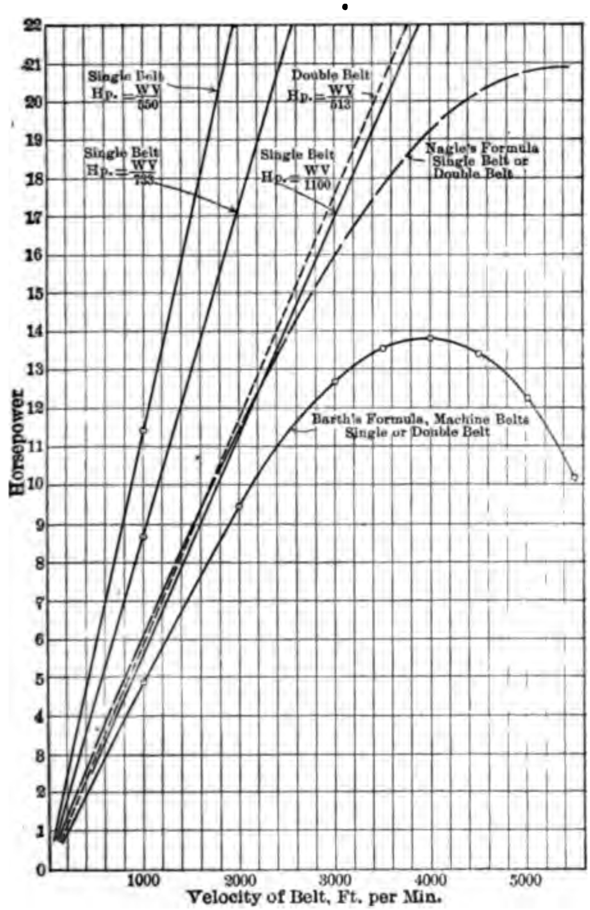


FIG. 1.—COMPARISON OF VARIOUS FORMULÆ FOR HORSEPOWER TRANSMITTED BY BELTING.

according to the different formulæ which have been discussed on pages 3 and 4. It will be observed that the rule of thumb formulæ give straight-line curves, while the Barth and

Nagle curves show the effect of centrifugal tension at the higher velocities. It will be observed that under the Barth practice a belt will transmit considerably less power at a given velocity than it will according to any of the other formulæ. It may be argued that for a given horsepower this will call for a heavier belt than it has heretofore been customary to use. This is precisely what it is intended to do. The object of the Barth practice is not to find the smallest belt that can be used for a particular service, but rather to find that belt which under the conditions imposed will transmit the required power at the least expense, taking into consideration the first cost of the belt, the expense of repairs and maintenance and the loss which would occur as a result of belt breakages in working hours. To accomplish this object low tensions are necessary, and low tensions mean large belts.

A belt put on the pulleys under a certain working tension will not maintain that tension for any length of time. The tension will constantly decrease and in time will fall below the point at which it will transmit the desired horsepower. It must, therefore, be retightened at intervals to restore it to its original initial tension. The interval between retightenings will vary with the age of the belt. A new belt will stretch rapidly, and probably will require taking up within forty-eight hours. The second retightening may be necessary three or four days later, and the third within a week after that. The intervals progressively increase in length until the period that will elapse between retightenings reaches three or four months. They then will remain practically constant. No belt, however, should be allowed to go for a longer period than six months without having its tension measured and being shortened to restore it to its original tension.

In connection with their investigations into the subject of belting, Messrs. Taylor and Barth devised a system of belt inspection and maintenance which would ensure that all belts in a plant would receive attention, have their tensions meas-

ured and be taken up, that is, shortened in order to tighten them, at such intervals as would prevent their stretching to a point where they would fail to do the work demanded of them. It also furnishes insurance that the belt will not break while in use, and thus stop production. This feature of the belting problem is every bit as important as correct original design, and the necessity of regular inspection and maintenance cannot be too strongly impressed upon the factory manager. The details of the system are set forth in Chapter V, devoted to belt maintenance.

The practical conclusions of Taylor and the mathematical deductions of Barth, which have resulted in the system of practice set forth in this book, have been amply justified by the results attained in the many factories that have adopted this practice. Notwithstanding that the first cost of the belts may be higher than it was under the older systems, or lack of system, the total cost of belting measured over a period of years has been far below that which obtained when the practice of light belts and high tensions was followed.

CHAPTER II

TAYLOR'S INVESTIGATIONS ON BELTING

The basis of modern belt practice is the nine-year experiment conducted by Frederick W. Taylor on belts in the machine shop of the Midvale Steel Company. The experiment was described by him in a paper before the American Society of Mechanical Engineers [1] in 1893. This investigation was radically different in its character and purpose from those which had preceded it. The usual experiments were of short duration and their chief object apparently had been to ascertain the value of the coefficient of friction. Taylor's object was to acquire information as to the cost and maintenance of belts, the cost of interruptions to manufacture due to belt failures, and information as to tensions, treatment, etc., that would give the lowest total cost for belting under operating conditions.

The opportunity for these experiments was afforded by the erection of a new machine shop in which many of the machines from the old shop were to be used. In the new shop the tight and loose pulleys on countershafts were made of larger diameter and wider than formerly, permitting the transmission of about two and one-half times the former belt power from main line to countershafts. The belts used were endless, joints being made by splicing, glueing and pegging instead of lacing or hooking, and double belts were used throughout the shop. By means of belt clamps having spring balances between the two pair of clamps, the tension to which the belt was subjected was accurately weighed when the belt was put on the pulleys and each time it was tightened. Provision was made for tightening belts by raising and lower-

[1] Transactions, American Society of Mechanical Engineers, vol. xv, p. 204.

ing the countershafts by the interposition of wooden packing pieces under the bearing.

Initial Tension. Each belt was spliced so that it was under an initial strain with the belt at rest immediately after tightening, of 71 lb. per inch of width of double belts. This is equivalent in the case of oak-tanned and fulled belts to 192 lb. per square inch of cross-section; and oak tanned, not fulled belts to 229 lb. per square inch of section; semi-rawhide belts 253 lb. per square inch of section; rawhide belts 284 lb. per square inch of section.

The belts were divided into two general classes known respectively as shifting and cone belts. The shifting belts were those running from the main line of shafting to the tight and loose pulleys on the countershaft, while cone belts extended from the countershaft to the machine. The shifting belts had about 2½ times as much transmitting power as the cone belts, which latter were used according to the ordinary rules of belting, such as are illustrated in the straight-line formulæ, Fig. 1, page 7. The table, pages 12–13, is a summary of the results of the nine-year experiment as obtained from a study of the records. The shifting belts were run, on an average, twenty hours per day, while the cone belts did not receive quite such steady use. While the shifting belts had a net working load only four-tenths as great as that of the cone belts, both shifting and cone belts were tightened repeatedly to the same tension, namely, 71 lb. per inch of width. The cone belts were supposed to give an effective pull on the pulley of 65 lb. per inch of width, and the shifting belts an effective pull of 26 lb. per inch of width.

Sizes and Speeds of Belts. The shifting belts ranged in dimensions from 39 ft. 7 in. long, 3½ in. wide and ¼ in. thick to 51 ft. 5 in. long, 6½ in. wide and 0.37 in. thick. The cone belts ranged from 24 ft. 7 in. long, 2 in. wide, ¼ in. thick to 31 ft. 10 in. long, 4 in. wide and 0.37 in. thick. The speed of the shifting belts ranged from 565 to 1570 ft. per minute

SUMMARY OF RESULTS OF EXPERIMENT IN LEATHER BELTING LASTING NINE YEARS.

Item No.		Shifting Belts.				Cone Belts.				
		Oak-tanned and fulled.	Semi-rawhide.	Rawhide.	Average of all shifting belts.	Average of all cone belts.	Oak-tanned and fulled.	Oak-tanned, not fulled.	Semi-rawhide.	Rawhide.
26	Average length of belts in feet	49	50	54	50	29	30	28	29	30
27	Average width of belts in inches	5	5	4.50	4.84	3.8	3.82	3.31	4.5	3.75
28	Average thickness of belts in hundredths of an inch	0.37	0.28	0.25	0.31	0.31	0.37	0.31	0.28	0.25
29	Average speed of belts in feet per minute				1140	580				
30	Percentage of belts still in use after 9 yrs. use (20 hrs. per day)*	All*	All	80	93.33*	27*	57	None	25	20
31	Percentage of belts worn out during 9 years	None	None	20	6.66	73	43	All	75	80
32	Average original cost per belt in dollars	36.44	46.17	43.53	40.16	18.04	16.34	13.53	22.42	21.88
33	Average cost per belt of cleaning, greasing, tightening, and repairing, in dollars	9.80	14.66	14.86	12.23	27	24.27	19.18	31.18	36.05
34	Average ratio that care and maintenance bears to first cost, %	0.27	0.31	0.34	0.304	1.50	1.49	1.42	1.39	1.65
35	Average cost per belt of cleaning and greasing, in dollars	4.41	4.50	3.74	4.24	2.56	2.68	2.64	2.41	2.40
36	Average cost per belt of tightening, in dollars	1.90	2.31	2.69	2.24	4.64	4.31	3.51	5.23	5.83
37	Average cost of belt of repairing, in dollars	3.37	7.81	8.42	5.75	19.88	17.27	13.03	23.53	27.90
38	Average cost of belt grease used in dressing belts	10 cents per belt per year.								
39	Average cost of belt glue used in splicing and repairing	13 cents per belt per year.								
40	Average number of years belts lasted†	9	9	8½	8.8*	6.7†	7.2	5.5†	8.5†	5.8†
41	Average cost in dollars per belt per year of service, including original cost and cost of maintenance and repairs	5.13	6.76	6.87	5.70	6.72	5.64	5.95	6.30	9.98
42	Ratio that total cost per year of life of belt, as given in Item 41, bears to first cost of belt, %	14	14.6	15.8	14.2	37.3	34.2	44	28	45.4
43	Total stretch of belts in inches as entered in record‡	11.9	16.9	17.3	15‡	21.4‡	16.7	21	21.4	28
44	Ratio of total stretch to total length of belt, %‡	.02	.028	.027	.023‡	.06‡	.046	.062	.066	.078
45	Average number of times each belt required tightening‡	2.66	3	3.6	3‡	14.5‡	12	12	16	20
46	Tension in lbs. per inch of width to which belts were strained each time that they required tightening	71	71	71	71	71	71	71	71	71
47	Tension in lbs. per square inch of section of belt to which belts were strained each time that they required tightening	192	253	284	239	239	192	229	253	284
48	Average strain in lbs. per inch of width of belt to which the tension of belts had fallen when they required tightening	22.7	21.5	19.3	21.1	32.9	37.6	30	29.6	34.3
49	Average strain in lbs. per square inch of section of belt to which tension of belts had fallen when they required tightening	61	77	77	68	106	102	97	106	133

Item No.		Shifting Belts. Oak-tanned and fulled.	Semi-rawhide.	Rawhide.	Average of all shifting belts.	Cone Belts. Average of all cone belts.	Oak-tanned, and fulled.	Oak-tanned, not fulled.	Semi-rawhide.	*Rawhide.
50	Average net (estimated) working load in lbs. per inch of width of belt. This is the effective pull per inch of width which the belts were supposed to give on the rim of the pulley	26	26	26	26	65	65	65	65	65
51	Av. net (estimated) working load in lbs. per sq. in. section of belt	70	93	104	84	210	175	210	232	260
52	(Estimated) average tension in lbs. per inch of width of belt exclusive of working load	32	26.5	28.5	28	46	49	44	43	47
53	(Estimated) average tension in lbs. per square inch section of belt	86.4	95	114	90	151	133	142	154	188
54	Average total (estimated) load (including working load and tension) in lbs. per inch of width	58	52.5	54.5	54	111	114	109	108	102
55	Average total load in lbs. per square inch section of belt (estimated)	157	188	218	174	358	308	351	385	408
56	Average number of months from the time that belts were tightened to a tension of 71 lbs' per inch of width until they required re-tightening	32	12.9	21	22	2.52	4.41	1.30	1.66	2.71
57	Average number of inches that each belt stretched from the time it was tightened until it next required tightening	4.48	5.31	5.56	5.11	1.55	1.52	1.82	1.43	1.45
58	Ratio of stretch between tightenings to total length of belt, %	0.73	0.83	0.87	0.81	0.47	0.41	0.61	0.40	0.47
59	Average stretch per belt in inches that takes place in first six months	None rec.	3.2	3.3	2.2	7.8	3.08	7.8	6.7	14
60	Percentage of the total stretch of belt that takes place during the first six months‡		19	19	14.6‡	36‡	18.4	37	31	50.6
61	Average number of times each belt was tightened or repaired	5.5	9.27	10	6.6	32	29	26	37	40
62	Average number of times per year of life of belt that it was either tightened or repaired	0.6 times	1.03	1.18	0.75	5times†	4	4.7†	4.4†	7†

* An examination of the shifting belts still in use shows them to be practically about as good as new, while the cone belts still in use are nearly worn out.

† Among the cone belts, two oak-tanned not fulled, five semi-raw-hide, and one raw-hide belt stretched unevenly, and were returned to their makers a few months after being put in use, and these were not taken into consideration in the averages of Item 40, although they were considered in averaging Item 30. None of the oak-tanned and fulled belts stretched unevenly.

‡ I do not consider the data regarding total stretch of cone belts as at all reliable. For the first few years the records of stretch were accurately kept, but when the belts began to require repairs, this portion of the record was carelessly kept, since frequently the belts (see Item 45) were tightened at the same time that they were repaired· the total stretch of the cone belts (see Item 43) certainly exceeded that given in the table. The data regarding the stretch of belts given in Items 57, 58, and 59 is reliable, as only instances concerning which there was no question were considered in making up the averages.

and the range of speed of the cone belts was from 225 to 1340 ft. per minute.

Factors which Influence Life of Belts. A study of the table on pages 12–13 will show an immense superiority of the shifting belts over the cone belts in every item excepting the first cost. This superiority is greater than the figures indicate, because as a rule the cone belts in use at the time the table was published were nearly worn out, while the shifting belts were in as good condition as when they were put into service. Mr. Taylor drew the conclusion that the life of the shifting belts would be three times that of the cone belts, and at the time of the publication of the table the total cost of the shifting belts per year of service was lower than that of the cone belts.

According to Mr. Taylor, the items which chiefly affect the life and satisfactory running of belting are as follows:

1. The material from which belts are made and the method of their construction.
2. Means of fastening and tightening them on the pulleys; i.e., whether laced, spliced or fastened with hooks.
3. The care and regularity with which they are greased and whether they are kept clean and free from oil.
4. The general nature of the services which belts are called upon to perform.
5. Whether belts run vertically or horizontally.
6. The relative length of the belts.
7. The relative speed of the belts.
8. The tension under which they are tightened.
9. The average total load to which they are subjected while working.

The eighth and ninth elements undoubtedly are those which have the greatest influence upon the life of belts. While all belts, both shifting and cone, were tightened to the same initial tension of 71 lb. per inch of width or 239 lb. per square inch of section, the shifting belts were, as a result of

their higher speeds and greater width, subjected to a lighter load than the cone belts. To this effect is due undoubtedly their superiority over the cone belts. If the cone belts had run under as low tensions as the shifting belts, they would probably have run an equally long time without requiring retightening.

The cone belts would fall from the initial tension of 71 lb. per inch of width to 33 lb. in a period of two months, while the shifting belts ran for twenty-two months before they fell from the tension of 71 lb. to a tension of 21 lb. Furthermore, shifting belts with a light load stretched 0.81 of 1 per cent before they required retightening, while the cone belts under twice as great a load required retightening after they had stretched but 0.47 of 1 per cent. Summarizing the experiments, Mr. Taylor stated that the total life of belts, the cost of maintenance and repairs and of the interruptions to manufacture caused by belts are primarily dependent upon the total load to which they are subjected. The other factors having the greatest influence on the durability of the belt are: Whether or not the belts are spliced or fastened with lacing or hooks; whether they are properly greased and kept clean; and the speed at which they run.

Average Load on the Belts. The average total load on the belting in the experiments was 54 lb. per inch of width with shifting belts and 111 lb. per inch of width with cone belts. That these figures represented radical departures from the ordinary practice of the day is evident from a comparison of the figures given by authors and writers of that time. The safe load on belts per square inch of section, according to different writers, ranged from 290 to 500 lb. per square inch. These figures are based, as a rule, upon the tensile strength of the leather, an arbitrary percentage of this strength being taken as the safe load as suited the whim of the writer.

The 54 lb. per inch of width of double belts which repre-

sents the total load on shifting belts, gives a total load of 174 lb. per square inch of section. Belts running under this load have already been shown to run much more economically and satisfactorily than cone belts which ran under a total load of 111 lb. per inch of width or 358 lb. per square inch of section. Evidently then the most economical total load for belting must lie between 174 and 357 lb. per square inch of section.

Taylor's Rules for Economical Belting Practice. Basing his conclusions partly upon the experiments and partly upon arbitrary assumptions, Mr. Taylor formulated the following rules as representing the most economical practice:

1. The average total load on belting should be 200 to 225 lb. per square inch of cross-section of belt.
2. Six- and seven-ply rubber belts and all double leather belts except oak-tanned and fulled, will transmit economically a pull of 30 lb. per inch of width. Oak-tanned and fulled belts will transmit economically a pull of 35 lb. per inch of width.
3. The most economical speed of belting is between 4000 and 4500 ft. per minute.

The conclusions which Mr. Taylor drew from the series of experiments outlined above were chiefly as follows:

1. Belts are more durable and work more satisfactorily when made narrow and thick, rather than wide and thin. As belts increase in width they should also be made thicker. On pulleys 12 in. diameter or larger, double belts are advisable and triple belts on pulleys of 20 in. diameter or larger. Quadruple belts should be used where possible on pulleys 30 in. diameter or larger.

2. Double belts will last well when repeatedly tightened under a strain when at rest of 71 lb. per inch of width or 240 lb. per square inch of cross-section. They will not, however, maintain this tension for any length of time. The most economical average total load for double belting is from 65 to 73 lb. per inch of width; that is, 200 to 225 lb.

per square inch of section. This corresponds to an effective pulling power of 30 lb. per inch of width.

3. If double belts are tightened while at rest to 71 lb. per inch of width and subjected to an additional working load of 65 lb. per inch of width, the tension in $2\frac{1}{2}$ months will fall to 33 lb. per inch of width while at rest or 106 lb. per square inch of cross-sectional area. The average tension during this period will be 46 lb. per inch of width and the average total load 111 lb. per inch of width, or 358 lb. per square inch of section. When double belts are first tightened to 71 lb. per inch of width and then subjected to an additional working load of 26 lb. per inch of width, their tension will fall in twenty-two months to 21 lb. per inch of width or 68 lb. per square inch of section. The average tension during this period will be 28 lb. per inch of width or 90 lb. per square inch of cross-section, and the average total load will be 54 lb. per inch of width and 174 lb. per square inch of section.

4. The total stretch of leather belting will exceed 6 per cent of the original length. The stretch during the first six months of the life of a belt is 36 per cent of the entire stretch if the belt is a double belt working under a total load of 111 lb. per inch of width and giving an effective pull of 65 lb. per inch of width. If, however, the belts are working according to the more economical rule, under a total load of 54 lb. per inch of width, giving an effective pull of 26 lb., the stretch during the first six months will be but 15 per cent of the entire stretch. A double belt will stretch 0.47 per cent of its length before requiring to be tightened when subjected, according to the ordinary rules, to a total load of 111 lb. per inch of width. A double belt with a total load of 54 lb. per inch of width and an effective pull of 26 lb. stretches 0.81 per cent before requiring retightening.

5. Belt clamps having spring balances between the two pair of clamps should be used for weighing the tension of the belt accurately, each time it is tightened. When it is

impractical to weigh the tension of a belt in tightening it, a safe rule is to shorten a double belt one-half inch for every 10 ft. of length if it requires tightening when working under a load of 111 lb. per inch of width. If working under a total load of 54 lb. per inch of width, the belt may be shortened one inch for every 10 ft. of length when retightened.

6. The total life of belting, cost of maintenance and repairs and the interruptions to manufacture caused by belts, are dependent upon the total load to which the belts are subjected more than upon any other condition. The method of fastening and the speed are the other chief considerations affecting their life. Double belts treated with care and running night and day at a moderate speed, should last for seven years when under a total average load of 111 lb. per inch of width. They should last for eighteen years when under an average total load of 54 lb. per inch of width. The speed at which belting runs has comparatively little effect on its life until it is in excess of from 2500 to 3000 ft. per minute. The belt speed for maximum economy should be from 4000 to 4500 ft. per minute.

7. In the machine shop in which the experiments were made, the average cost per double belt per year of service, including the first cost and the cost of maintenance and repairs, proved to be $6.72 for belts used under a total load of 111 lb. per inch of width. For belts under a total load of 54 lb. per inch of width the cost was found to be less than $5.70. The cost of all labor and materials used in the maintenance and repairs of double belts plus the cost of renewals as belts give out through a term of years, will amount to an average of 37 per cent per year of the original cost of the belts if belts are tightened accurately so that they will work under an average total load of 111 lb. per inch. If, however, they are worked under a total load of 54 lb. per inch of width, the annual cost of maintenance, repairs and renewals amounts to not more than 14 per cent of the first cost and probably

less. The cost of maintenance and repairs of double belts during a total life of 6.7 years, running night and day, amounts to $1\frac{1}{2}$ times the first cost of the belt when under a total load of 54 lb. per inch of width. The cost through a term of 8.8 years amounts only to 30.4 per cent of the total cost.

8. The best distance from center to center of shafts is from 20 to 25 ft. The faces of the pulleys should be about 25 per cent wider than their belts. When practicable, belts should be tightened by moving one pulley away from the other. Countershafts should be mounted on frames and raised in order to tighten vertical and diagonal belts.

9. When it is necessary to run night and day throughout the week and work without stopping, each important belt should be supplied with an idler pulley, which can be tightened upon it while running in case of slipping. Idler pulleys work most satisfactorily when located on the slack side of the belt about one-quarter of the distance from the driving pulley.

10. Belts should be cleaned and greased every five to six months.

11. Belts of any width can be successfully shifted backward and forward on tight and loose pulleys to throw lines of shafting in and out of use. The best form of belt shifter for wide belts is a pair of rollers twice the width of the belt, either of which can be pressed against the flat surface of the belt on its slack side close to the driving pulley, the axis of the roller making an angle of 75 degrees with the center line of the belt.

Comments on Taylor's Rules. Commenting on Mr. Taylor's conclusions, William Kent [1] has compared the rules for belting given by Mr. Taylor with some of the common rules of thumb of the form H.P. $= WV \div c$ adopted by practical mechanics. The practical mechanic very often figured that a single belt one inch wide, running at a rate of 1000 ft.

[1] Transactions, American Society of Mechanical Engineers, vol. xv, p. 242.

per min., that is $c = 1000$, would transmit one horsepower; whereas, Mr. Taylor's rules indicate the use of a velocity constant of 950 to 1100 for a double belt. Assuming that a double belt is twice as strong or will carry twice the power of a single belt, Taylor's rule calls for a belt almost twice as large as was common practice at the time of the experiments. This would result in a first cost for belting double that called for by even the most liberal of the then existing rules. Inasmuch as the Taylor practice was aimed at finding the belt which would transmit a given horsepower at the least cost for maintenance, repair and interruptions to manufacture rather than the smallest belt which would transmit a given horsepower, the rules are on totally different planes. The question might be raised as to why the Taylor rule using 1100 for a velocity factor for a double belt is correct when it has been found possible to obtain such good results from a belt proportioned with a velocity factor of only 550 for a single belt; that is, a belt only one-quarter as heavy as a belt called for by the Taylor rule. The answer is that in many cases belts designed under the old rule are really running under the Taylor rule. For instance: A belt proportioned for and estimated to be driving four horsepower is usually driving only one horsepower. A belt is usually, in mechanics' or millwrights' practice, made wide enough to drive a machine at the maximum power guessed to be required at any time. If this amount of power was required to be transmitted continually, the belt would probably need tightening once a month, and would be worn out within two years. Having to transmit the maximum power only occasionally for brief intervals and averaging a power only one-quarter of the maximum, it may require retightening only once a year and would last for twenty years. It would, therefore, be used as an eminent example of the merits of the old rule, and give rise to conclusions which would be wholly incorrect.

CHAPTER III

THE HORSEPOWER TRANSMITTED BY LEATHER BELTING

General Formula for Horsepower of Belting. The general formula for the horsepower transmitted by a leather belt of 1 sq. in. cross-sectional area is

$$\text{H.P.} = \frac{pV}{33000}, \qquad (1)$$

in which p is the effective pull in the belt in pounds, and V is the velocity of the belt in feet per minute. If we let t_1 and t_2 represent respectively the tensions in the tight and slack strands of a running belt,

$$p = t_1 - t_2. \qquad (2)$$

The determination of the value of p is by no means the simple proposition that it appears from the formula, nor is it necessary for those who have no particular liking for mathematical problems, and who are interested only in the amount of horsepower that a belt will transmit under a given set of conditions, such as are usual in the shop, or who desire to ascertain the size of belt that should be used to transmit a given amount of power to a machine or group of machines. For this purpose the tables on pages 91 to 110 are available, and will prove of greater practical value than the mathematics by which they were calculated. The use of the tables will, therefore, be explained, and the mathematics of them postponed to another chapter. They can be ignored by those not interested. The tables are calculated from the formulæ developed by Carl G. Barth and represent the practice recommended by him in his paper "The Transmission of Power by Leather Belting." [1]

[1] Transactions, American Society of Mechanical Engineers, vol. xxxi, p. 29.

The tables show the horsepower that will be transmitted under any given set of conditions, and also the tensions under which the belts should be put up, and the minimum tensions to which they may be allowed to fall in service before they should be retightened to the original tension. If the belts

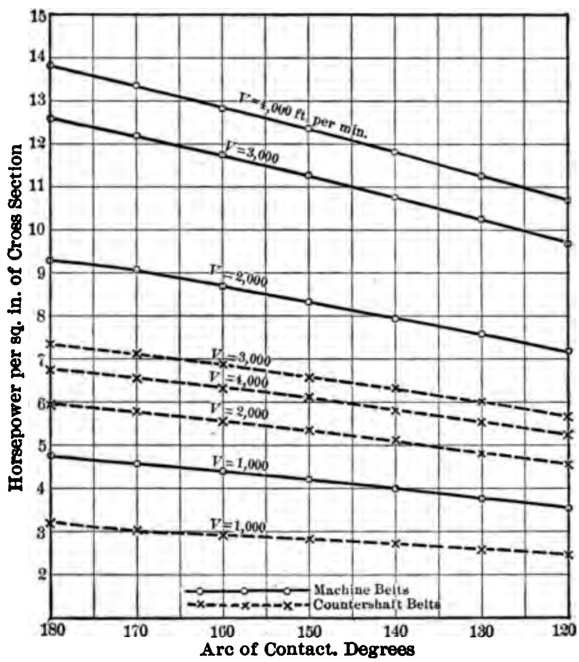


FIG. 2.—RELATION BETWEEN ARC OF CONTACT AND HORSEPOWER TRANSMITTED BY MACHINE AND COUNTERSHAFT BELTS.

are put up under the tensions given in the tables, while they are at rest, they will when running develop in the tight and loose strands such values of t_1 and t_2 as will give the value of p, which will at the velocity that the belt is supposed to attain develop the required horsepower.

Effect of Arc of Contact. The arc of contact of the belt on the pulley has a direct bearing on the horsepower that can

be transmitted and it should be determined first. The arc on the smaller pulley is the one that should be considered, as the horsepower transmitted is limited by it. If the smaller pulley is the driver, it limits the amount of power that can be delivered to the belt, while if it is the driven, the arc limits the amount of power that the belt can deliver to the pulley. The curves, Fig. 2, show the influence of arc of contact upon horsepower transmitted. The tables on pages 96 to 101 indicate the arc of contact of the belt on the smaller of a pair of pulleys for the usual range of pulley sizes and center-line distances. The arc that the belt subtends on the larger pulley can be found by subtracting the figure given in the table from 360 degrees.

Effect of Velocity of Belt. Equally or more important than arc of contact is the velocity of the belt. With a given velocity, up to the point at which centrifugal tension begins to manifest itself, the horsepower transmitted varies directly with the velocity. As explained in Chapter I, the effect of centrifugal tension reaches serious proportions in the neighborhood of 4000 ft. per minute. This is clearly shown by the curves, Fig. 3, which illustrate the relation between velocity and horsepower transmitted. The tables on pages 102 to 107 give the belt velocity obtained by a varying number of revolutions per minute of pulleys of different diameters within the usual range of shop sizes.

Use of Horsepower Tables. Having determined the arc of contact of the belt on the smaller pulley and the velocity, the horsepower that the belt will transmit per square inch of cross-sectional area can be read directly from the proper section of the horsepower table on pages 91 to 94. In the third and ninth columns of the table will be found the tensions per square inch of cross-section under which the belt should be put on the pulleys; the fourth and tenth columns show the tension below which it should not be allowed to fall in service, and the point at which the belt should be retightened to the

tensions given in columns 3 and 9. The use of the tables presupposes that the belts will be subjected to regular inspection and measurement of the tension, according to the system outlined in Chapter V, or a system that will accom-

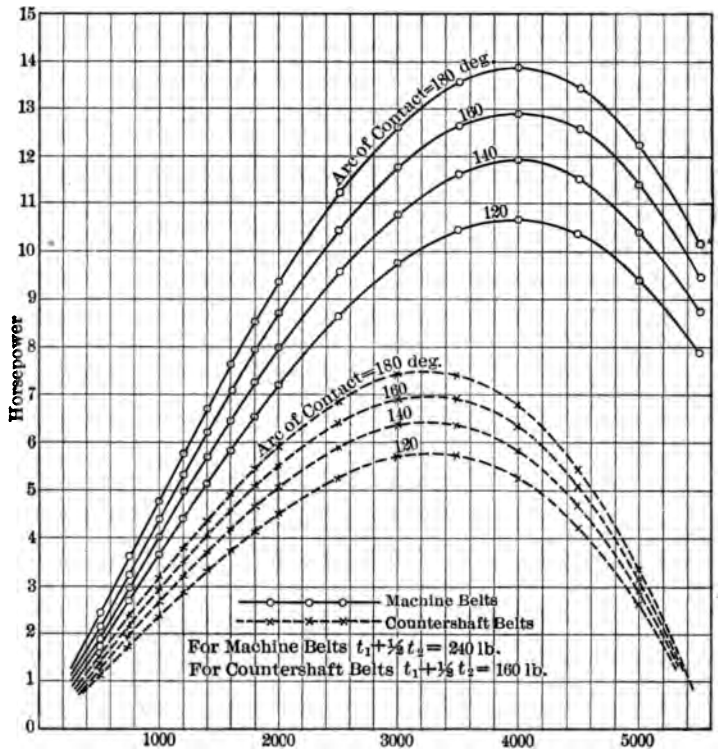


FIG. 3.—RELATION BETWEEN VELOCITY OF BELT AND HORSEPOWER TRANSMITTED PER SQUARE INCH OF BELT CROSS SECTION.

Horsepower Computed by Barth's Formulæ.

plish the same result, to insure that the tension of the belt will not fall, while in service, below the minimum tension as given in the tables. While the tables are calculated for belts of 1 sq. in. of cross-sectional area, they can be adapted to belts of any width and thickness by means of the table of factors on

page 108, which shows the relative cross-sectional area of belts of different sizes. As the horsepower transmitted under a given set of conditions varies directly with the cross-sectional area of the belt, a simple multiplication of the figure obtained from the horsepower table by the constant in the table of relative areas will give the desired result. The same statement applies to the tension values. Multiplying the initial and minimum tension values as given in the horsepower tables by the constant from the table of relative areas will give the total tension under which any belt should be put up, and the total tension below which it should not be allowed to fall in service and at which it should be shortened and retightened.

Examples in the Use of the Tables. The following examples will make clear the use of the tables:

Example I. The three steps of a lathe cone pulley have diameters respectively of 14, 16 and 18 in. The face of each step is 5 in. The countershaft cone has the same diameters, and rotates at a speed of 110 r.p.m. The distance between the center lines of the machine spindle and the countershaft is 8 ft. What size of belt is necessary to deliver 3 horsepower to the machine cone when the belt is on the largest step of that cone? What horsepower will be delivered to it when the belt is on the smallest step of the machine cone? What will be the maximum and minimum initial tension in the belt under which it should be put up, and below which it should not be allowed to fall in service?

The difference between the diameter of the largest step of the machine cone and the smallest step of the countershaft cone is 4 in. Referring to the arc of contact table on page 96, we find that for a 4-in. difference in diameter, a center-line distance of 8 ft. will give an arc of contact on the smaller pulley of 178 degrees. For all practical purposes this can be regarded as 180 degrees. The velocity table on page 103 shows that a 14-in. pulley at 110 r.p.m. will give a

belt velocity of 403.2 ft. per minute. Turning now to the horsepower table on page 91 we find in the section headed "180 Degrees Arc of Contact" under machine belts, that a belt of 1 sq. in. of cross-sectional area will transmit 1.05 horsepower at 250 ft. per minute and 2.27 horsepower at 500 ft. per minute. The horsepower that is transmitted at speeds intermediate to those in the table increases very nearly in proportion to the increase in speed. The difference in speed between 250 and 403 ft. per minute is 153 ft. per minute, and between 250 and 500 ft. per minute is 250 ft. per minute. The difference in horsepower transmitted at 250 ft. and 500 ft. per minute is, according to the table, 1.22 horsepower. Therefore, at 403 ft. per minute the horsepower transmitted would be

$$\begin{aligned} \text{H.P.} &= 1.05+\left(1.22\times\frac{153}{250}\right) \\ &= 1.05+1.22\times 0.612 \\ &= 1.79. \end{aligned}$$

As it is desired that the belt shall be capable of transmitting 3 horsepower, the cross-sectional area of the belt will be $3 \div 1.79 = 1.676$ sq. in. The width of belt that should be used on a cone pulley of 5-in. face should not exceed $4\frac{1}{2}$ in. We therefore look in the table of relative areas on page 108 opposite the width of $4\frac{1}{2}$ in. for the figure most nearly corresponding to 1.676. We find it to be 1.719 under the thickness of $\frac{3}{8}$ in. Therefore a belt $4\frac{1}{2}\times\frac{3}{8}$ in. will be ample to transmit 3 horsepower under the conditions. It should be put on the pulleys under a total initial tension (see column 3 of table) of

$$\begin{aligned} &1.719\times\left\{183.75-\left(183.75-180.25\times\frac{153}{250}\right)\right\} \\ &\qquad = 1.719\times 181.61 = 312 \text{ lb. approximately.} \end{aligned}$$

By the same sort of a calculation we find from column

of the table that the belt should be taken down and retightened when its total tension has fallen to 215 lb.

To find the horsepower that the belt will transmit when it is on the smallest step of the machine cone, we ascertain from the velocity table its velocity, which in this case is that of an 18-in. pulley at 110 r.p.m. or 518.3 ft. per minute. This can be considered as 500 ft. per minute. The arc of contact will be the same as before, and as before can be taken as 180 degrees. We may then read the horsepower directly from the horsepower table as 2.27 horsepower per square inch of cross-sectional area of belt. Multiplying this figure by the area of the belt in question, 1.719, we obtain as the horsepower transmitted 3.9.

Example II. A 25-horsepower motor is located in the roof trusses of a machine shop and drives the shafting for a group of machines. The motor has a speed of 1100 r.p.m., the motor pulley being 9 in. diameter and 9 in. face. The motor is frequently called on to carry an overload of 10 per cent, or a total load of 27.5 horsepower. The center line distance between the motor and the line shaft is 13 ft. 6 in. The pulley on the line shaft is 30 in. diameter. The present belt is 8 in. wide and $\frac{7}{16}$ in. thick. Is it heavy enough for the loads, and if not, what size of belt should be used? At what tensions should the belt be put up and retightened?

As the belt is in an inaccessible position, the figures in that portion of the horsepower table headed "Countershaft Belts" will apply. From the table of arcs of contact we find the arc of the belt on the smaller of the two pulleys to be 172 degrees, and the answers to our problem will therefore be found in that section of the horsepower table entitled "170 Degrees." From the velocity table the velocity of the belt is ascertained to be 2591.3 ft. per minute. Inasmuch as the arc of contact is actually larger than 170 degrees, we can assume the velocity to be slightly less than it really is and read the horsepower directly from the table as 6.66,

opposite the velocity of 2500 ft. per minute, for a belt of 1 sq. in. of cross-sectional area. The total power to be transmitted is 27.5 horsepower and therefore the area of the belt to be used is $27.5 \div 6.66 = 4.129$ sq. in. In the table of relative areas we find under $\frac{7}{16}$ in. thickness of belt, the value of 3.5 opposite the width of 8 in. The present $8 \times \frac{7}{16}$ in. belt is too small for the work it is required to do if it is to be run under the most economical conditions. If its use is persisted in it will be necessary to operate it at a higher tension and to tighten it at more frequent intervals. If we desire to use the same width of belt, which is quite as wide as should be used on a pulley of 9 in. face, we must increase its thickness. The table of relative areas shows that an 8-in. belt $\frac{1}{2}$ in. thick will have a cross-sectional area of 4 sq. in. and that an $8 \times \frac{9}{16}$-in. belt will have an area of 4.5 sq. in. As the former will transmit $6.66 \times 4 = 26.64$ horsepower it can safely be used without increasing the tensions given in the horsepower table to any great extent. It might be advisable to tighten the belt at a slightly higher figure than that given in the table as the minimum tension. Assuming the $8 \times \frac{1}{2}$-in. belt to be chosen, the total tension under which it should be put on the pulleys will be $4 \times 142 = 568$ lb., and the tension at which it should be retightened $4 \times 85.5 = 342$ lb., or to be on the safe side, 350 lb. total tension.

CHAPTER IV

THEORY OF TRANSMITTING POWER BY BELTING

IN developing the theory and formulæ on which the horsepower and tension tables on pages 91 to 94 are based, Mr. Barth took as his starting-point the investigations of Frederick W. Taylor, which are discussed in Chapter II. In considering these formulæ and in using the tables which are calculated from them, certain facts must clearly be kept in mind: 1. The pulling power of a belt depends, among other things, upon the tension in the belt when it is at rest, called, in what follows, the initial tension. Upon this initial tension depends, in part, the tensions in the tight and slack strands of the running belt; that is, the values t_1 and t_2 in formula (2), in the previous chapter. 2. That a belt in service stretches and thereby decreases the initial tension. This stretch is rapid at first and decreases in amount with the age and use of the belt. 3. That a belt whose initial tension is allowed to fall below a predetermined figure will, at a given velocity of belt, fail to transmit the desired horsepower. 4. That excessive unit tension in a belt will cause it to stretch rapidly and will increase the maintenance and other charges relating to the belting. 5. As a corollary to the foregoing, heavy belts, at moderate tensions will give, all things considered, better service than light belts at high tensions. 6. From the standpoint of belt maintenance, it is desirable to equalize as far as possible the intervals during which belts running at different velocities and transmitting different horsepowers will fall to the minimum allowable unit tension and require retightening. 7. That belts that are readily accessible, such as machine belts, can be taken down and retightened more easily than can belts in inaccessible loca-

tions, such as countershaft and jackshaft belts, and therefore can be run at high tensions. This explains the reasons for presenting two sets of figures in the horsepower tables for machine and countershaft belts respectively. 8. An initial unit tension of 200 lb. per square inch is about the maximum that should be put upon a belt if it is to be used under the most economical conditions.

Before taking up the theory enunciated by Mr. Barth we will examine the general theory of transmitting power by means of belting.

General Theory of Power Transmission by Belting. A belt one inch wide is stretched on a pulley as in Fig. 4, by

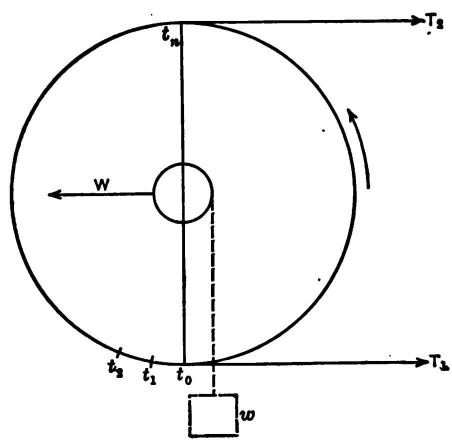


FIG. 4.—DIAGRAM ILLUSTRATING THEORY OF BELTING.

two equal forces T_1 and T_2. The pull on the shaft is $T_1+T_2 = W$. If R is the radius of the pulley in inches, the radial or normal pressure P brought on the pulley at each inch of length of contact is $\frac{T_1}{R}$, assuming that the belt is elastic and free to move around the pulley without friction. Now assume that there is friction between the pulley and the

belt and that T_1 is increased and T_2 is decreased so as to cause rotation of the pulley. The difference of the tensions, $T_1 - T_2$, is represented in Fig. 4 by a weight being lifted as the pulley is rotated. With given values of T_1 and T_2 the tensions in the belt will gradually decrease from T_1 at the point where it first comes in contact with the pulley to T_2 at the point t_n where it leaves the pulley. The decrease in tension in successive inches of length of contact may be represented by the quantities $(t_0 - t_1)$, $(t_2 - t_1)$, etc., and the sum of these differences will equal the total difference in tension $T_1 - T_2$, which is the same as the effective pull or tractive force, p.

With a given sum of the tensions $T_1 + T_2 = W$, the limit to the tractive force p, or to the weight that can be lifted without serious or objectionable slipping of the belt, depends upon the length, α, of the belt in contact with the pulley expressed in radians, and upon the coefficient of friction f. The relation existing between the four quantities T_1, T_2, f and α is expressed by the exponential formula

$$\frac{T_1}{T_2} = e^{f\alpha}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

This equation is derived by means of the calculus, and those interested in the mathematical discussion involved are referred to Church's "Mechanics of Engineering," Rankine's "Machinery and Millwork" and other works on applied mechanics.

Coefficient of Friction. The coefficient of friction is defined as the ratio of the force required to cause one surface to slide upon another to the pressure with which the surfaces are in contact. In the case of belting it is evident from an inspection of the formula $\frac{T_1}{T_2} = e^{f\alpha}$, that the influence that the coefficient of friction exerts on the power transmitted by a belt is very great. The value given by experimenters from the time of Morin up to the experiments of Lewis and Ban-

croft ranged all the way from 0.15 to 1.38. The coefficient of friction was determined in the early experiments by hanging a section of a belt over a pulley or drum, suspending weights from the two strands and noting the difference in weight at which the belt would begin to slip. This method is correct enough for determining the coefficient of friction of leather on iron or wood under certain limited conditions, but it manifestly does not represent the conditions under which belts are operated. In the first place the coefficient is determined at practically zero velocity. Secondly, this method takes no account of the alternate lengthening and shortening that takes place in the slack and tight strands of the belt as it passes over the pulleys, giving rise to the phenomenon of belt creep. These factors have a distinct effect on the coefficient of friction and cannot be neglected.

The experiments of Wilfred Lewis [1] showed that the value of the coefficient of friction depended in part on the velocity of slip of the belt on the pulleys, and that other conditions influencing it were the condition of the leather, the temperature and the pressure. Mr. Lewis's experiments indicated that a total allowance of 2 per cent for the slip of the belt on a pair of pulleys was good practice; that is, a slip of 2 per cent will not be injurious to the belt nor to the service that it is performing. This allowance includes the creep of the belt as described below, but which was not considered separately by him.

The phenomenon of belt creep was studied by Prof. W. W. Bird, who presented his conclusions,[2] in 1895. Belt creep was explained by Prof. Bird as follows:

Referring to Fig. 5, *A* is the driving and *B* the driven pulley, running in the direction indicated by the arrow. The tensions in the tight and slack sides are T_1 and T_2 respectively. One inch of slack belt goes on the pulley *B*

---

[1] Transactions, American Society of Mechanical Engineers, vol. vii, p. 549.

[2] *Ibid.*, vol. xxvi, p. 584.

at o, and at or before the point p it feels the effect of the increased tension and stretches to $1+s$ inches. It travels from p to m and goes on pulley A while stretched. At or before reaching point n, as the tension decreases it contracts to one inch and so completes the cycle. With a light load the belt creeps ahead of the pulley B at or near the point p. If the load is heavy the creep works toward the point o and the belt may slip. This also takes place when the belt tensions are too light, even with small loads. On the pulley A the belt creeps back relative to the pulley surface at m, and as in B may extend to the point m where regular slip will occur.

Prof. Bird derived the following formulæ as the result

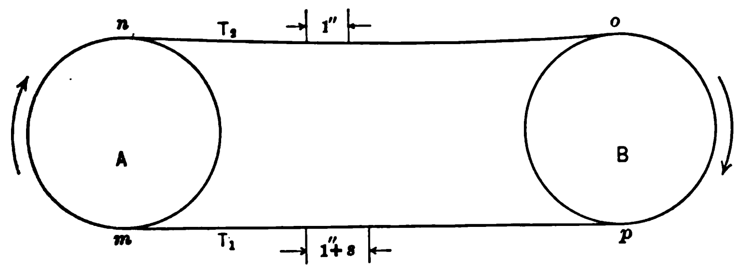


FIG. 5.—DIAGRAMMATIC REPRESENTATION OF THE PHENOMENON OF BELT CREEP.

of his experiments, x being the percentage of slip, v_1 and v_2 respectively the pitch velocities of pulleys A and B, p the effective tension (T_1-T_2), pound per square inch, E the modulus of elasticity of the leather, and s the stretch per inch due to p.

$$x=\frac{v_1-v_2}{v_1};\quad \frac{v_1}{v_2}=\frac{1+s}{1};\quad \frac{E}{p}=\frac{1}{s};\quad x=\frac{s}{1+s}=\frac{p}{E+p}.$$

The table on page 34 shows the percentage of creep for different values of p and a range of values of E.

The value of $E=20{,}000$ was stated to be a fair value for ordinary working conditions. Prof. Bird concluded that

Percentage of Belt Creep for Various Values of p and E

p	E			
	5000	10,000	15,000	20,000
25	0.49	0.25	0.17	0.12
100	1.96	.99	.66	.50
125	2.44	1.23	.83	.62
150	2.91	1.48	.99	.74
175	3.38	1.72	1.15	.87

for common leather belt under ordinary working conditions the creep should be not over 1 per cent. While this is sometimes called legitimate slip, it is an actual loss of power and cannot be avoided by belt tighteners and patent pulley coverings.

A study of the experiments of Messrs. Lewis and Bird by Mr. Barth led to the development of an equation for the coefficient of friction f as follows:

$$f=0.6-\frac{2}{4+v},$$

in which v is the total average sliding velocity of the belt in feet per minute, it being based on belts in active service and tested without the application of belt dressing. The equation covers in a satisfactory manner results obtained with belts in service.

To be of practical use, however, the velocity of sliding of the belt should in some way be connected with its pulling power at a given actual velocity of belt. This is accomplished through the medium of the ratio of the effective tensions expressed in formula (1), namely $\frac{T_1}{T_2}=e^{f\alpha}$, which ratio will later be used in determining a value for p. It is quite evident that an average coefficient of friction for all velocities of belt, such as was assumed in the rule of thumb formulæ, and even in more pretentious ones, would be incorrect, in view of the work of Messrs. Lewis and Bird. Mr. Barth further maintains that basing the coefficient of friction on an average

total sliding velocity of belt corresponding to a fixed percentage of the belt speed would also be incorrect. The reason for this is that even a very low percentage would mean a high sliding velocity with a high-speed belt, while a high percentage would mean only a moderate sliding velocity in the case of a low-speed belt. For example, a slip of 1 per cent in the case of a belt velocity of 4000 ft. per minute would be an actual slip of 40 ft. per minute, while a 5 per cent slip at a belt velocity of 300 ft. per minute would be but 15 ft. per minute of actual slip.

The studies all pointed to the fact that the coefficient of friction is a quantity that varies with the velocity, and its value was finally fixed by Mr. Barth, after considerable study, by the empirical formula

$$f=0.54-\frac{140}{500+V}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

in which V is the velocity of the belt in feet per minute.

Further studies of Mr. Lewis's work convinced Mr. Barth that it was best, due to conflicting results, to leave out of consideration entirely the effect of initial tension on the coefficient of friction, and to ignore the conclusion that it depends in part on the intensity of the pressure. The values of f according to the above formula for velocities ranging from 0 to 6500 ft. per minute, are given in the table on page 38.

Effect of Centrifugal Force. A heavy particle moving in a curved path of radius R exerts a force on the arm, cord or other medium which constrains it to move in the curved path. This force is termed centrifugal force and is represented by the formula

$$f=\frac{Wv^2}{gR},$$

in which W is the weight of the body in pounds, V the linear velocity in feet per second, R the radius, in feet, of the path

through which the body moves, and g is the acceleration due to gravity.

A belt passing around a pulley develops this centrifugal force which appears as a tension in the belt whose effect is to diminish the tensions in both the tight and slack strands. If T_c be the tension due to centrifugal force, in pounds per square inch of section of belt, then the tension in the tight and slack strands will be $(T_1 - T_c)$ and $(T_2 - T_c)$ respectively. In the formula on page 35 W is the weight of the belt on

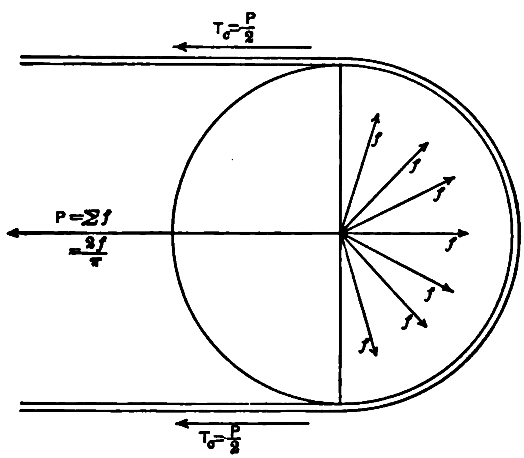


FIG. 6.—ACTION OF CENTRIFUGAL FORCE.

the semi-circumference of a pulley. If w be the weight of 1 cu. in. of belting, then for a strip of belting 1 in. wide and 1 in. thick

$$W = 12 w \pi R.$$

If v is the velocity of the belt in feet per minute, then $V = \frac{v}{60}$ and $V^2 = \frac{v^2}{3600}$. Substituting these values in the general formula we have

$$F = \frac{12\pi R w \frac{v^2}{3600}}{gR} = \frac{\pi w v^2}{300 g}.$$

The centrifugal force in the belt acts radially around the entire circumference, and is balanced by the tension in the two strands of the belt at the extremes of the diameter, as diagrammatically represented in Fig. 6. Each unit of length of the belt will have developed in it a centrifugal force of magnitude f, and acting in the direction shown by the radial arrows. The sum of these unit forces will be the total centrifugal force in the belt, that is $\Sigma f = F$. The horizontal components of the various forces f are balanced by a force P acting in the opposite direction and which manifests itself as a tension in the belt at the points a and b. The tension T_c in each strand is $\frac{1}{2}P$. The magnitude of the sum of these horizontal components is the sum of the forces f multiplied by the ratio of the diameter to the length of the semi-circumference along which they act, that is by $2/\pi$. Therefore

$$P = 2T_c = \frac{2F}{\pi} = \frac{2}{\pi} \frac{\pi w v^2}{300g} = \frac{2wv^2}{300g}.$$

Hence

$$T_c = \frac{wv^2}{300g}.$$

If the weight of a cubic inch of leather belting be taken as $\frac{1}{30}$ lb. and g be put equal to $32\frac{1}{6}$ we have

$$T_c = 0.000003454v^2, \quad \ldots \ldots \quad (3)$$

which is the formula given by Barth and which agrees substantially with those given by Nagle and Rankine.

Barth's Theory of Belting. In developing the theory of belting forming the basis of practice in the most progressive shops, which theory accords with the seven requirements set forth in the first paragraph of this chapter, Mr. Barth found that certain assumptions, formerly held to be true by writers on belting, were, as a matter of fact, fallacious, and he therefore discarded them.

The two most important assumptions thus found to be untrue were: (1) The sum of the tensions in the two strands

of a running belt is constant under any variations in the effective pull. (2) The coefficient of friction between the belt and its pulley is constant for any given set of conditions, irrespective of the velocity of the belt. In place of the first of these two erroneous suppositions, Mr. Barth enunciated and proved the following theorem: *Under any variation in the effective pull of a belt, the sum of the square roots of the tensions in the two strands remains constant.*

Coefficient of Friction. To take the place of the theory of a constant coefficient of friction, Mr. Barth derived the empirical formula (2), given on page 35, which gives an increased value to the coefficient with increasing velocity of the belt. The formula gives a value which ranges from 0.26 at zero velocity to 0.52 at a belt velocity of 6500 ft. per minute. The value of the coefficient of friction as given by Morin in his experiments was, for oily belts 0.15; for greasy belts, 0.23; for wet belts, 0.36, and for dry belts, 0.56. Messrs. Towne and Briggs in their experiments in 1868 recommended the value of 0.42 as expressing an average of the conditions of belt operation, while Releaux recommended 0.25. The values of the coefficient of friction, according to formula (2), are given in the table below:

VALUES OF THE COEFFICIENT OF FRICTION OF LEATHER BELTS IN IRON PULLEYS. (BARTH'S FORMULA.)

Velocity of Belt, Ft. per Min.	Coefficient of friction *f*.	Velocity of Belt, Ft. per Min.	Coefficient of friction *f*.	Velocity of Belt, Ft. per Min.	Coefficient of friction *f*.
0	0.260	800	0.432	3000	0.500
50	0.285	900	0.440	3500	0.505
100	0.307	1000	0.446	4000	0.509
200	0.340	1200	0.458	4500	0.512
300	0.365	1400	0.466	5000	0.514
400	0.384	1600	0.473	5500	0.517
500	0.400	1800	0.479	6000	0.519
600	0.413	2000	0.484	6500	0.520
700	0.423	2500	0.493		

It will be observed that these values are within the range given by the experimenters whose work is noted above. While the formula is not claimed to give values that are exact

for every condition of the surfaces of the belt and pulley—no empirical formula could do this—yet it does give values which are perhaps very close to the true ones, when the belts are maintained in accordance with the practice recommended by Taylor which is outlined in Chapter II, and they agree with the results obtained in practice.

Working Tensions. The basic rule of practice adopted by Mr. Barth is as follows: For the driving belt of a machine, the initial tension must be such that when the belt is doing the maximum amount of work intended, the sum of the tension on the tight side of the belt and one-half the tension on the slack side will equal 240 lb. per square inch of cross-section for belt speeds; and for a belt driving a countershaft or one in any other inaccessible location, this sum will equal 160 lb. The maximum initial tension, that is, the tension under which the belt is put upon the pulley when it is placed in service and the tension to which it is to be retightened as often as it falls to the minimum, must be such that the above derived sum is 320 lb. for machine belts and 240 lb. for countershaft belts. These values represent the constant *A* in the mathematical discussion which follows and upon which are based the horsepower tables on pages 91 to 94.

With the above facts in mind, we can now examine the formulæ upon which the tables of horsepower and tension are based. The complete mathematical discussion of their development is contained in Mr. Barth's paper, "The Transmission of Power by Leather Belting," heretofore referred to, and is there available for those who wish to follow it. Only the briefest possible explanation will be given here.

NOTATION

t_1 = tension in the tight strand of the running belt, pounds per square inch.

t_2 = tension in the slack strand of the running belt, pounds per square inch.

t_0 = minimum initial tension in the belt at rest on its pulleys, pounds per square inch. This is the tension to which the belt can be allowed to fall before retightening.

t_m = maximum initial tension in the belt at rest upon its pulleys, pounds per square inch. This is the tension to which the belt is retightened.

t_c = centrifugal tension in belt, or loss in effective tension due to centrifugal force, pounds per square inch.

p = effective pull in belt, pounds per square inch.

α = arc of contact of the belt on the smaller pulley, in radians $= (\pi/180) \times$ arc in degrees.

f = coefficient of friction between the belt and the pulleys.

V = velocity of belt, feet per minute.

e = basis of Naperian system of logarithms $= 2.71828$.

A = a constant; whose values range from 160 to 320 as explained below.

The problem is to find a formula which will express the relation of the initial tensions t_0 and t_m to the tensions t_1 and t_2 in the tight and slack strands of the belt, in order to ascertain what value must be given to t_0 to obtain a predetermined value of p in the formula (4):

$$p = t_1 - t_2. \qquad (4)$$

Bearing in mind the conditions under which the intervals at which the belts should be retightened will be made equal as explained in a previous paragraph, we can express this relation algebraically as

$$t_1 + \tfrac{1}{2} t_2 = A. \qquad (5)$$

For machine belts $A = 240$ and 320, depending upon whether values of t_0 or t_m respectively are being sought, as explained later; for countershaft belts $A = 160$ and 240 respectively for finding values of t_0 and t_m.

The passage of the belt around the pulleys develops in it centrifugal force whose effect is to decrease the tension

in either strand of the belt; that is, the effective tension in the tight strand is (t_1-t_c) and in the slack strand (t_2-t_c). The total friction of the belt on the pulley is the ratio of the effective tensions. It has previously been shown that this ratio which is the tractive or driving force of the belt on the pulley, is

$$\frac{t_1-t_c}{t_2-t_c}=e^{f\alpha}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

In order to make use of the common system of logarithms, instead of the Naperian system, and to obtain values of α in degrees instead of radians, both of which will be found more convenient, the expression $10^{0.00758f\theta}$ may be substituted for the expression $e^{f\alpha}$, θ being taken in degrees. That is, the ratio of the effective tensions is the number whose common logarithm is $0.00758f\theta$.

To obtain values of f, we may use the empirical formula

$$f=0.54-\frac{140}{500+V}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The centrifugal tension developed at any velocity we have already found to be

$$t_c=0.000003454V^2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

While a formula has been developed for obtaining directly the value of t_1, it is here only of academic interest, because of the fact that we can, by combining formulæ (2), (3), (4), (5) and (6), obtain a formula expressing the value of p in terms of f, α, A and V. As the value of p must in any case be determined, and as from it the values of t_1 and t_2 can then be ascertained, if desired, by much simpler formulæ it is unnecessary to consider here the fundamental formula for t_1. Combining the above mentioned formulæ we obtain

$$p=\frac{(e^{f\alpha}-1)\,(2A-0.00001036V^2)}{2e^{f\alpha}+1}. \quad . \quad . \quad . \quad (7)$$

which can be converted, by the substitution of the expression given above for $e^{f\alpha}$ to

$$p=\frac{(10^{0.00758f\theta}-1)(2A-0.00001036V^2)}{2\times 10^{0.00758f\theta}+1} \quad . \quad . \quad . \quad (8)$$

Having determined a value for p, the horsepower can readily be determined from the general formula

$$\text{H.P.}=\frac{pV}{33000}.$$

It now only remains to connect the formula for p with the initial tension in the belt as it is at rest on the pulleys. This connection is established by the formula

$$t_{\text{initial}}=\frac{4A-p+\sqrt{(4A-p)^2-9p^2}}{12}. \quad . \quad . \quad . \quad (9)$$

This formula will give values for t_0 and t_m respectively, according as we substitute for A, 240 or 320 for machine belts, and 160 or 240 for countershaft belts. The above are the formulæ on which the horsepower tables on pages 91 to 94 are based.

If, after having determined the values of p, t_0 and t_m, t_1 and t_2 are desired, the following formulæ may be used:

$$t_1=\frac{2A+p}{3}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$t_2=\frac{2(A-p)}{3}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Formulæ (10) and (11) may be used if it is desired to determine the initial tension in a horizontal belt, which theoretically is slightly different from that which should be used for a vertical belt, to which formula (9) applies. This tension, however, differs so little from that of a vertical belt, except in the case of belts of extraordinary length, that there is no practical end served by determining it separately. The formula, however, is given here simply as a matter of interest: For long horizontal belts

$$t_{\text{initial}}=\frac{1}{4}\left[\sqrt{t_1}+\sqrt{t_2}-\left(\frac{C}{5}\,\frac{t_1-t_2}{t_1t_2}\right)^2\right]^2, \quad . \quad . \quad (12)$$

in which C is the center line distance of the pulleys in inches, the notation otherwise being as before. In making use of formula (12) p is first determined by formula (7) or (8), and then t_1 and t_2 are evaluated by formulæ (10) and (11).

The use of the formulæ is simple. The number of separate computations involved in determining the values of p, t_0 t_m and the horsepower for a given velocity, however, is large, and for a large number of belts the operation would be exceedingly tedious, if not prohibitive from the standpoint of time alone. To facilitate the application of the formulæ to belts which fall within the usual shop range, Mr. Barth has constructed the slide rule illustrated in Fig. 7. This rule embodies all the above formulæ except (12), and by means of it, practically every problem in belt driving that may arise in the ordinary factory can be solved. The tables on pages 91 to 110 embody the same information, and may to some prove of easier application.

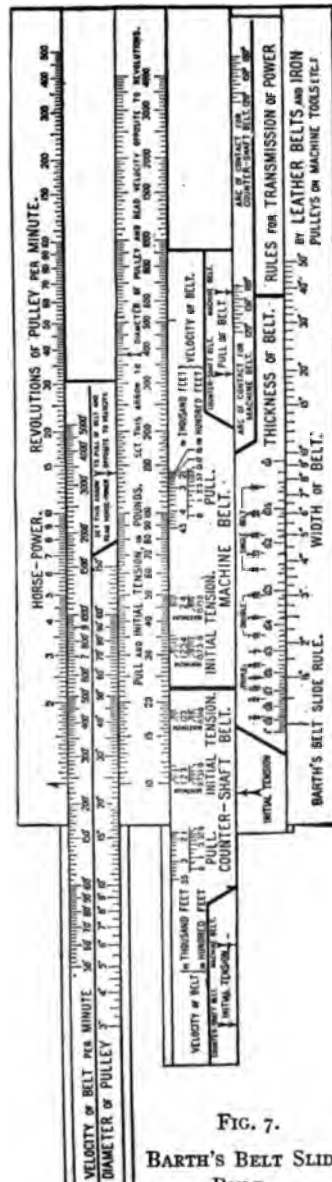


FIG. 7.
BARTH'S BELT SLIDE RULE.

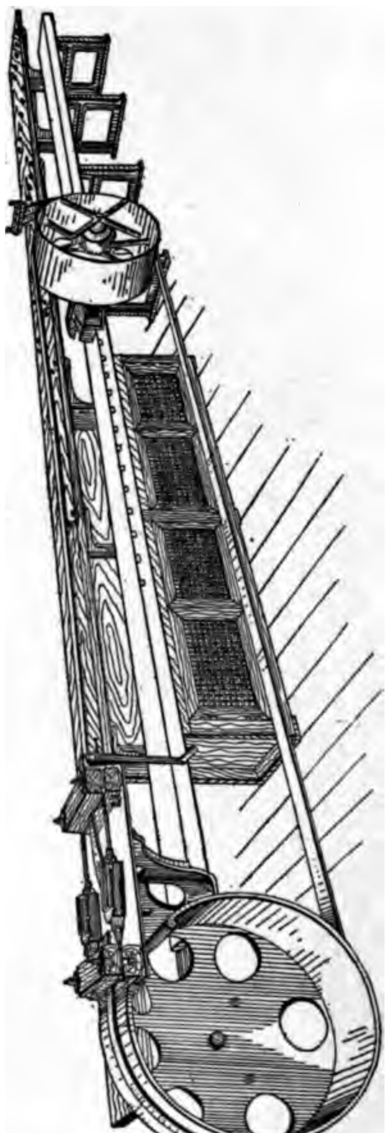


FIG. 8.—BELT BENCH FOR MEASURING TENSION IN BELTS.

CHAPTER V

BELT MAINTENANCE

In the previous chapters, stress has been laid on the necessity of putting a belt on the pulleys under a certain predetermined tension, depending upon the horsepower it is to transmit and the velocity at which it is to be operated, and of retightening it to this tension whenever it falls to a predetermined minimum. While it is possible to weigh the tension in the belt at intervals while it is directly on the pulleys, it is inconvenient to do so, and the " belt bench " forms a useful and convenient appliance for insuring that belts are kept up and maintained at the proper tension.

The Belt Bench. A form of belt bench which has been widely used is shown in Fig. 8. It is made by the Tabor Manufacturing Co., of Philadelphia. This bench consists of a channel which forms a runway for a carriage on which is mounted a movable drum. The channel is supported upon cast-iron chairs and brackets and a bench top is placed over the channel. A fixed drum is at one end of the channel. The channel is graduated to indicate distances of 1 ft. in the length of the belt, while the carriage carrying the movable drum is graduated in inches and fractions of an inch of length of belting, as shown in Fig. 9. The length of the belt needed for a given pair of pulleys is measured by means of a steel tape passed around the pulleys, and the movable drum on the carriage of the belt bench is set at the graduation indicated by the reading of the tape. The carriage having been set to the length of the belt required, the belt is passed around the drums and the two ends are passed through belt clamps and held firmly therein. The belt is set at the same distance from the edge of each drum, by means of

graduations on the drums. The two sets of clamps are connected by means of a pair of spring balances which are attached to a pair of screws, as shown in the illustration, Fig. 10.

By means of a spindle and miter gears acting on the nuts at the end of the screws, the two screws on the balance can be drawn up uniformly by rotating the handle at the end

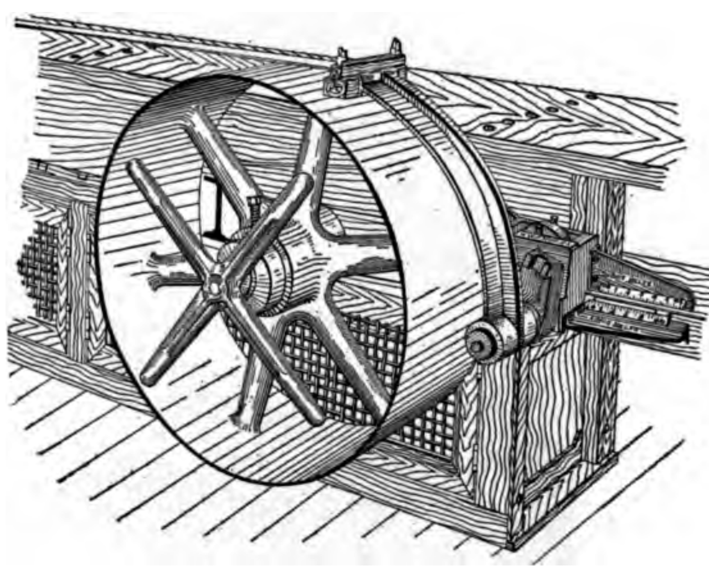


FIG. 9.—CARRIAGE OF BELT BENCH.
Clamping Arm in Position to Apply Tension to a Short Belt.

of the spindle. As the clamps are drawn together by means of the screws, the tension in the belt is registered on the scales of the spring balance. The belt is drawn up until the tension registered is the maximum initial tension under which the belt should be kept in service, as indicated by the slide rule or the tables on page 91. The ends of the belt are then cut off square and it is removed from the belt bench, laced and put on the pulleys. The same procedure is followed in determining the tension in a belt which has been

in service. The belt is taken down from the pulleys, and the belt bench having been set to the length of the belt, the clamps are put on and the belt drawn up until the ends come together. The tension is then read, and if it is at or below the minimum to which the belt should be allowed to fall in service, it is cut off to a length sufficient to restore in it once more the maximum initial tension. The chart, Fig. 32, on page 95, is for use in connection with the tables of horse-

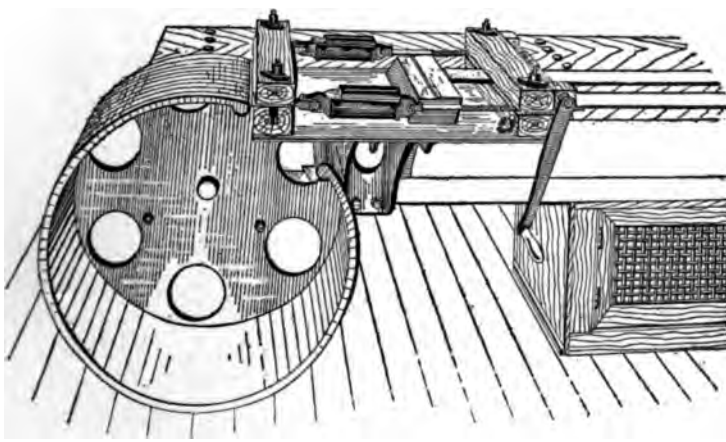


FIG. 10.—HEAD END OF BELT BENCH.
When Applying Tension to Short Belt.

power and tension. It shows the relative stretch of belts under different tensions.

The above procedure applies to belts in excess of 16 ft. in length. If the belts are under 16 ft. long, they are not passed around the drum as described, but one end is gripped in an auxiliary clamp on the movable carriage as shown in Fig. 9. The carriage is adjusted by means of the upper scale seen at the right of the drum. The other end of the belt is gripped in one of the belt clamps, the other clamp being attached to the bench as shown in Fig. 10. The tension is then weighed by means of the scales as before.

Belt Maintenance. In order that belts in service shall not fall below the minimum tension, it is essential that they be inspected and their tensions weighed at such intervals as will insure their being retightened before they have stretched to a point where they will fail to render the service required of them. To insure this inspection a system of maintenance is absolutely essential. The following system is condensed from instructions issued by the Tabor Manufacturing Co. in connection with the belt benches above described.

Belt Data Necessary. The following data are necessary in the operation of the system and should be ascertained for every belt in the shop:

1. Location of the belt.
2. Its purpose; that is, whether it be used for a cone or machine drive, countershaft drive or feed drive, etc.
3. Exact length of the belt over the pulleys as taken with a steel tape.
4. Width of the belt.
5. Thickness of the belt.
6. Maximum velocity at which the belt can be operated.
7. Minimum velocity at which the belt can be operated.

These data are entered upon a belting record, Fig. 11, together with such other information as may be useful, such as the name of the maker, kind of belt and other information which is called for on the record. In addition, the maximum and minimum tensions as determined by the slide rule or the horsepower tables on page 91 should be entered.

Belt Symbols. For convenience in identifying belts, it is desirable to provide a code of symbols, the use of which will completely describe and locate any belt in the shop without the necessity of using long written descriptions. The following code of symbols is that recommended by the Tabor Mfg. Co.:

GENERAL SYMBOLS

A, Auxiliary belts: *i. e.*, all belts from countershaft to the machine used for driving auxiliary parts. These may be subdivided as follows:

AO, Open auxiliary machine belt.

AC, Crossed machine belt.

The machine number is prefixed in each case. Thus, the belt for traversing the table of grinding machine No. 16, would be indicated as 16 AO.

C, Countershaft belts. That is, all belts from the main line or jackshaft to countershafts. These belts can be subdivided in the case of a two-speed countershaft, as fast and slow belts and also as open and crossed belts giving rise to the following symbols:

CFO, Fast belt from line shaft to countershaft, open.

CSO, Slow belt from line shaft to countershaft, open.

CFC, Fast belt from line shaft to countershaft, crossed.

CSC, Slow belt from line shaft to countershaft, crossed.

CRO, Reverse belt from line shaft to countershaft, open.

CRC, Reverse belt from line shaft to countershaft, crossed. The machine number is prefixed in each case.

E, All belts from engine or motor to main line shaft or jackshaft. These may be subdivided into open and crossed belts, as follows:

EO, Open belts from engine or motor to jackshaft or main line shaft.

EC, Crossed engine or motor belts to main line or jackshaft.

If more than one engine or motor is in use in the shop the number of it is to be prefixed to the belt symbol.

F, Feed belts. Subdivided into

FO, Feed belts, open.

FC, Feed belts, crossed.

The machine number is prefixed in each case.

J, Jackshaft belts. That is, all belts from line shaft to the jackshaft. Subdivided into

JO, Jackshaft belts, open.

JC, Jackshaft belts, crossed.

All jackshafts should be numbered and the number of the jackshaft prefixed to the belt symbol.

L, Main line shaft belts. That is, all belts from engine jackshaft to the line shaft. Subdivided into

LO, Main line shaft belts, open.

LC, Main line shaft belts, crossed.

M. Machine belts. That is, all belts from countershafts to the machines and used for operating them. Subdivided into

MO, Machine belts, open.

MC, Machine belts, crossed.

S, All belts from main line shaft to side line shaft.

SO, Side line shaft belts, open.

SC, Side line shaft belts, crossed.

The various shafts should be numbered and a list of these numbers posted where it is accessible to the belt-fixer, showing the location of all shafts and machines.

Maintaining the Tension in Belts. New belts usually stretch to such an extent that they require tightening within twenty-four hours after being put up. The second tightening, as a rule, should take place forty-eight hours later; the third tightening at the end of a week, and another at the end of a month. Machine or cone belts should then be taken down and have their tension measured every two months, while for countershaft belts three to four months may elapse. The time at which any given belt should be tightened is determined by the entries made on the belting record, shown in Fig. 11.

For instance: Referring to the chart illustrated it will be noted that the belt was to be put on the pulleys under a tension of 130 lb. on each spring balance, or a total of 260 lb.

BELTING

DEPARTMENT D9n

BELT SYMBOL 126-CSO

LOCATION 30" Lodge + Shipley Lathe
PURPOSE Slow Countershaft belt
KIND Heavy Double belt
MAKER'S NAME Schieren
DATE PUT IN USE 5-31-11
DATE TAKEN OUT
LIFE OF BELT

LENGTH OVER PULLEYS 37'-6¾"
WIDTH 5"
THICKNESS 3/8"
CROSS SECTION IN SQ. IN. 1.875
HOW FASTENED Wire laced
BELT GREASE TO BE USED Oil + Tallow
BELT DRESSING TO BE USED Plomo

MAX. VELOCITY 1300'
MIN. VELOCITY 600'
MAX. TENSION 130# } ON EACH SPRING BALANCE
MIN. TENSION 76# }

COST OF BELT $33.15
COST OF MAINTENANCE
TOTAL COST

DATE RE-TIGHTENED	TENSION IN LBS. INDICATED BY EACH SPRING BALANCE		AMOUNT TAKEN OUT	DATE TENSION AGAIN SHOULD BE TAKEN	DATE CLEANED AND GREASED	DATE TO BE CLEANED AND GREASED AGAIN	DATE DRESSING PUT ON DURING SERVICE	GENERAL REMARKS ABOUT WORK DONE ON BELT, CONDITION OF BELT, ETC.	WORKMAN'S NAME	MAN'S TIME REQUIRED TO DO WORK	COST OF WORK
	BEFORE TIGHTENING	AFTER TIGHTENING									
5-31-11		130		6-1-11				New Belt	Ski	5	10
6-1-11	100	100	0	6-3-11					Ski	3	06
6-3-11	70	130	1¼	6-7-11					Ski	4	08
6-7-11	80	128	1⅛	6-15-11					Ski	4	08
6-15-11	85	125	1⅛	7-1-11					Ski	5	10
7-1-11	65	130	1½	8-1-11					Ski	4	08
8-1-11	95	95	0	9-1-11					Ski	6	12
9-1-11	70	132	1¼	12-1-12					Ski	4	08
					11-1-11	5-1-12			Ski	2	04
12-1-11	85	130	1	4-1-12					Ski	5	10
4-1-12	68	125	1¼	10-1-12					Ski	4	08
					5-1-12	11-1-12			Ski	2	04
10-1-12	60	130	1¼	4-1-13					Ski	4	08

FIG. 11.—BELT RECORD.

The belt was put up new on May 31st and the tension entered in the record. The belt being new, an entry was made in the proper column showing that its tension should be measured on the first day of June, and a memorandum to that effect was placed in the tickler file, which will be described later. On the first day of June the tension of the belt was again measured and found to be 200 lb. or 100 lb. on each spring balance. As the minimum tension to which a belt can be allowed to fall would be 76 lb. on each balance. it was unnecessary to shorten the belt and it was accordingly put back on the pulleys and a memorandum entered for the tension to be measured forty-eight hours later, or on June 3d. On June 3d the tension was measured and found to have fallen to 70 lb. on each balance, or 6 lb. per balance less than the minimum. An inch and a quarter was accordingly removed from the belt, the amount being determined by using the chart Fig. 32 on page 95, bringing its tension again to 130 lb., and a memorandum was entered to again weigh the tension on the 7th of June, four days later. On the 7th of June the tension was found to have dropped to 80 lb. per balance, and $1\frac{1}{8}$ in. was removed, bringing the tension back to 128 lb. per balance. The performance of the belt from that point on is clearly shown by the record and no further explanation is deemed necessary.

Belt-fixer's Orders. Fig. 12 is a form of order issued to the belt-fixer. This carries on it information which the belt-fixer requires, including the symbol of the belt, which also gives its location; the length, which is the length around the pulleys as measured by the steel tape; the maximum and minimum tensions between which the belt should operate. It also contains spaces in which the belt-fixer enters information required from him for entry on the belt record. This order is issued when a new belt is to be put up or an old one retightened. After doing the work, the belt-fixer returns the order to the person who is in charge of belt maintenance, with the

information entered as to the tension in the belt before and after tightening, the amount by which the belt was shortened, and any other information that may be desired. This information is duly entered on the record, and a new order is made out and filed in the tickler file so that it will come out on the day that the record shows that the belt will next

DM. 12 (Date) IN OUT 12.00 m	ORDER NUMBER D m L
DEPARTMENT DAY RATE	MAN'S TIME

LENGTH OF BELT 37′–6¾″ BELT SYMBOL I36-CS.O

MAXIMUM TENSION 130# MINIMUM TENSION 76#

CLEANED AND GREASED GREASE USED

DRESSED WHILE IN USE DRESSING USED

AMOUNT TAKEN OUT LENGTH PUT IN

LENGTH OF SPLICE CEMENT USED

TENSION IN LBS. INDICATED BY EACH SPRING BALANCE. { BEFORE TIGHTENING / AFTER

WORKMAN'S NAME MAN'S NO.

ENTERED IN		
PAY SHEET	COST SHEET	BELT RECORD

DAY WORK TIME NOTE

FIG. 12.—BELT FIXER'S ORDER.

need attention. If the belt is a new one the order will be filed to make its appearance the following day, while if it is an old belt it will be filed in the tickler file to appear on a date several months distant. In each case the return of an order by the belt-fixer showing that the work has been done, requires the making out of a new order which will automatically make its appearance from the tickler file on the next date that the belt should be inspected, as shown by

the belting record and according to the schedule outlined on page 52.

Cleaning and Greasing. At periodical intervals, say six to eight months, the belt-fixer should, in addition to weighing the tension, clean and grease the belt with an approved dressing. A good dressing is as follows:

A. Boiled linseed oil 37 per cent by measure. Tallow, 30 per cent by measure.

B. Machine oil, 27 per cent by measure. Beeswax, 6 per cent by measure.

A and *B* are heated separately to 360 deg. and mixed while hot. Additional information regarding belt dressings, cleaning and greasing is given in Chapter VII.

Time at which Belt Work Should be Done. Belts should be tightened and repaired outside of working hours, unless there is a belt break-down or a machine is standing idle with no prospect of being used during the time that the belt will be under repair. It is recommended that the belt-fixer be given six belts to tighten each day; three being done during the noon hour and three after quitting time in the afternoon. The belt orders should, therefore, be divided into lots of six, each lot being dated for the date on which the belt is to be tightened, and the lots filed accordingly in the tickler file. If the number of belts is such that by dividing the belts into lots of six, a greater period than two months will elapse before all the belts in the shop are inspected and tightened, it will be necessary to increase the number tightened in any one day and provide assistance for the belt-fixer, if necessary. This system provides for considerable flexibility, and good judgment on the part of the one in charge may be exercised to advantage. As the conditions vary under which different belts of the same size work, it is not always possible to lay down fixed rules as outlined above, and the engineer in charge of the work should then use his judgment in regard to shortening the periods during which certain belts are run,

before retightening. It is essential that all the work on belts be done by a regular belt-fixer according to the instructions issued from the belting record. Machine hands should not, under any circumstances, be allowed to tighten or repair their own belts.

Tickler File. The tickler file consists of a series of envelopes or portfolios, one for each day of the year, which are contained in a suitable cabinet. A very satisfactory file may be made by obtaining envelopes of stiff, heavy manila paper, the envelopes being $8\frac{1}{2} \times 11$ in., and open on one end. They can be filed in an ordinary vertical file drawer, with heavy Bristol board spacers separating the various months, or the weeks in each month if such subdivision is found to be desirable. Memoranda of every description of matters that will require attention on some particular day in the future are marked with the date on which they should be attended to and are then filed in the portfolio of that date. Every morning the envelope or portfolio for that day is opened and the various memoranda are distributed to those concerned with them. On Saturdays and on the days preceding holidays the portfolio for the following day is also opened and the memoranda therein are attended to on that day or are transferred to the portfolio of the day following. The belt orders as they are made out are dated with the date on which the belt in question should next be examined, and the order is then filed in the portfolio of that date in the tickler file. It is evident that the usefulness of the tickler file is not confined to the subject of belt repairs alone, but that it can be extended to cover repairs of all kinds. In fact it will be found useful for almost every activity that may take place in the factory or office.

CHAPTER VI

METHODS OF FASTENING BELTS

JOINTS in belts are made generally by one of four processes, namely, cementing or glueing (endless belts), rawhide lacing, metal lacing, and wire hinge joints. Of these, the cemented or glued joint is probably the least troublesome in service, but it is difficult to readjust the tension in the belt unless the pulleys on which the belt runs are adjustable and can be moved apart as the belt stretches. Rawhide lacing has been and probably is the most common method of making joints in belts, but it is rapidly giving way to the wire hinge joint. Metallic lacing is but comparatively little used, and it is not especially suited to belts running on small diameter pulleys.

Endless Belts. In endless belts the length of the splice should not be less than 9 in. in the case of a double belt. If the belts are more than 9 in. wide the splice should be made equal in length to the width of the belt up to 18 in., which is the maximum length necessary for any splice. In making the splice, one end of the belt is beveled with a small block plane. Care should be taken to see that the lap runs in the same direction as the other laps in the belt. The position of the lap is then marked off on the other end of the belt and it is also beveled. The surfaces of the two laps are then scraped smooth.

After sizing the laps with thin belt glue, a thin coating of belt glue should be applied as rapidly as possible to the entire surface of both laps. The laps are then brought together and hammered all over with a broad-faced hammer to ensure contact throughout the splice, after which the joint is squeezed in a press or between handscrews for about fifteen minutes until the glue has set. Rubber-faced shoes

should be placed on either side of the joint before it is squeezed. After glueing the splice should be pegged with shoemaker's pegs $\frac{1}{2}$ in. apart.

The belt glue should be applied hot to the splices whenever possible, although it may sometimes be necessary to use a fast-setting cold glue. The Tabor Manufacturing Co. recommends a glue composed of

Page's liquid fish glue........2 parts (measure)
Russian liquid isinglass......1 part (measure)

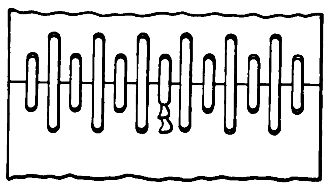
A belt cement recipe given in *Industrial Engineering*, October, 1913, calls for equal parts of glue and American isinglass, which are allowed to soak in water for ten hours. The mixture is then brought to the boiling-point and pure tannin is added until it has the appearance of the white of an egg. This glue is applied hot to the surfaces of the splice.

Endless belts should always be run so that the feather edge of the splice on the side next to the pulley points away from the pulley as the splice approaches it.

The main objection to endless belts is that the joints must be made with the belt in position, which is always inconvenient and often impossible. Furthermore there is no way of tightening them except altering the center line distance of the pulleys, unless the joint is broken and remade. Either method is troublesome and expensive, and therefore, despite their manifest advantages in other respects the endless belt is limited in its application. The common practice is therefore to make all the joints but one in the belt cemented joints, and to make the final joint a laced or wired one.

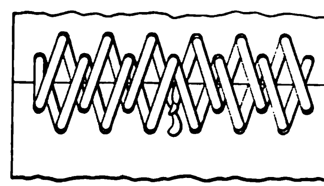
Laced Joints. The ordinary belt lacing is a strip of rawhide whose width depends on the size of the belt to be laced. The ends of the belt are cut square with the edges and butted together, after which holes are punched in the two ends with an oval punch, the large diameter of the oval being parallel with the sides of the belt. The lacing is started at the center,

worked over to one side of the belt, back to the opposite side, and is finally brought to the center at the point at which it started. The lacing is carried out according to the diagram, Figs. 13 and 14. The lacing is parallel to the sides of the belt on the face next to the pulley and crossed on the



Pulley Side

FIG. 13.



Opposite Side

FIG. 14.

METHOD OF LACING A BELT.

opposite side. The following table was published by A. D. Porter in the *Railway Age Gazette*, November 4, 1910, and shows the size of lacing and the punching of the holes for lacing various sizes of belts.

Width of Belt, in.	Punching of Lace Holes, Distance from Ends of Belt, in.		Distance of center line of first hole in first row from edge of belt, in.	Size of Belt, lace, in.
	First row.	Second row.		
2 to 4	3/8	3/4	3/8	3/16
6 " 8	1/2	1	1/2	1/4
10 " 12	5/8	1 1/4	5/8	5/16
14 " 16	3/4	1 1/2	3/4	3/8
18 " 20	7/8	1 3/4	7/8	7/16
22 " 24	1	2	1	1/2

The strength of a laced joint is about one-half that of the belt itself, or about 1000 to 1500 lb. per square inch. The working strain allowed on the driving side of the joint was generally taken by early authorities on belting as one-third of the ultimate strength, and this formed the basis of the tension in the belt for the old rules for horsepower. Inasmuch as the working tension in the belt recommended by later authorities, as Taylor and Barth, is considerably lower than

these figures, the strength of the joint has but little to do with the horsepower that can be transmitted.

The objections to the laced joint are that it is seldom well made, and that it is difficult to lace a belt so as to maintain the same tension in it on the two edges of the belt. It also virtually increases the diameter of the pulley at the point at which the joint is in contact with it, since the lacing cannot be made flush with the surface of the belt. The lacing causes the belt on either side of the joint to be raised

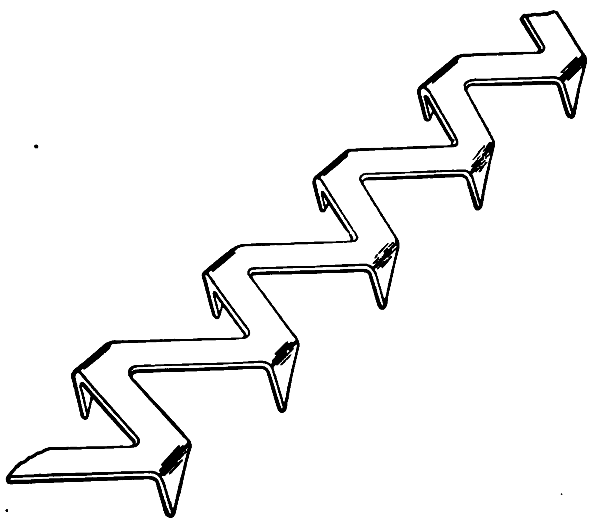


FIG. 15.—METALLIC BELT LACING.

from the pulley, and destroys the intimate contact of belt and pulley. The joint also is less flexible than the remainder of the belt and this has a more or less detrimental effect on the belt and the service rendered by it. Belts as a rule fail at or near the lacing. Added to the other objections, the making of a laced joint is slow and expensive.

Metallic Belt Lacing. A typical form of metallic belt lacing is the Bristol lacing shown in Fig. 15. This is of steel, and its construction is so clearly shown in the illustration that

no further description is necessary. The ends of the belt are butted together and a strip of the metallic lacing whose length is equal to the width of the belt is laid across the joint. The points are then driven through the belt with a hammer, the belt is turned over and the points are clinched

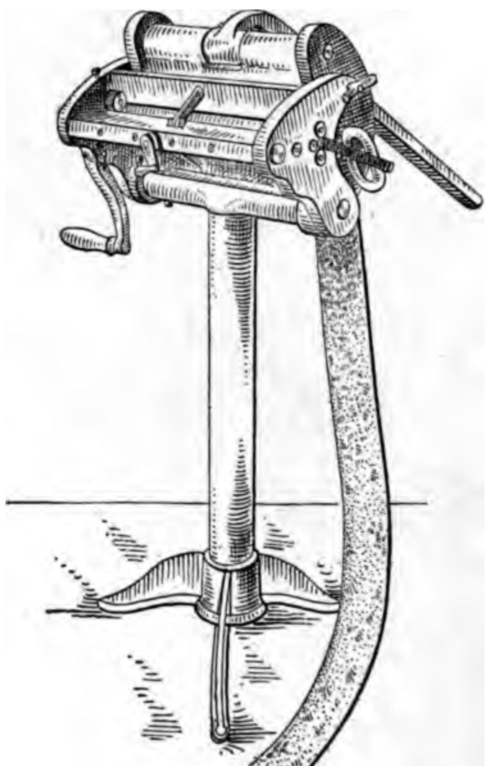


FIG. 16.—BELT LACING MACHINE.

on the pulley side, and driven into the belt so that there will be no projections to come in contact with the pulley as the joint passes over it. The metallic lacing provides a rapid and cheap means of making a joint, but it is less flexible than the rawhide lacing. It is also difficult to break the

joint for purposes of repair without seriously damaging the belt at the joint.

Wire-laced Hinge Joints. The most satisfactory method of making a belt joint, all things considered, is by the use of wire which is looped through each end of the belt in a series of eyelets which project beyond the end of the belt. The eyelets in the two ends are brought together and a pin of rawhide is run through them, thus joining the ends of the

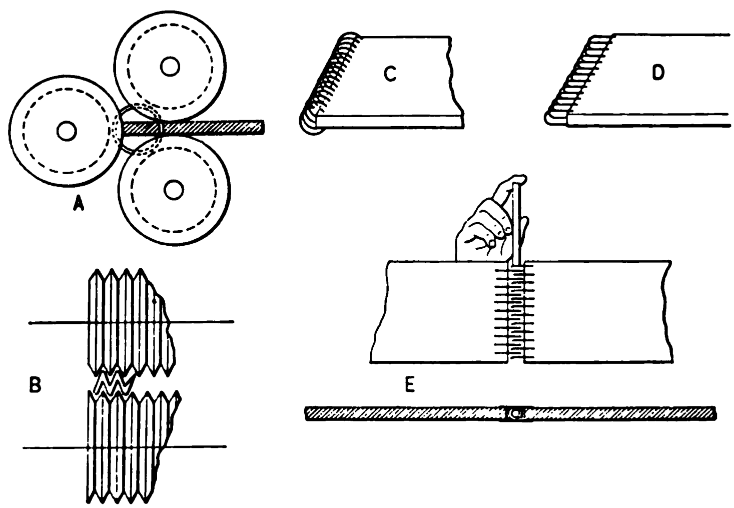


FIG. 17.—METHOD OF MAKING A WIRE-LACED HINGE JOINT.

belt and forming a true hinge. The finished joint does not project above the surface of the pulley, and as the joint can be bent through an arc of about 270 degrees there is no diminution in the flexibility of the belt as a whole. The joint is made in a small machine such as is shown in Fig. 16. Machines of this character are made by the Peerless Belt Lacing Machine Co., Philadelphia, and by the Birdsboro Foundry & Machine Co., Birdsboro, Pa.

The machine consists of three corrugated rolls arranged as shown at *A* in Fig. 17. The belt is inserted between the

two vertical rolls and against the third roll as shown. On revolving the crank, a helical needle inserted in the corrugations as shown at *B*, is carried through the end of the belt, making a series of holes across the belt. A coiled wire lacing is next inserted in the corrugations and by turning the crank is carried through the perforations in the belt as shown at *C*, Fig. 17. The coils are then flattened and forced into the belt as at *D*. The ends of the wire lacing should be bent back and driven into the belt so that they will not injure the hands when handling the belt. Each end of the belt is laced and the wire loops in the two ends are joined by means of a rawhide pin inserted between them as shown at *E*. This pin should not be allowed to project beyond the edges of the belt, but should be cut off about $\frac{1}{8}$ in. short on each side. The strength of a joint made in the above fashion is stated by the makers of the machine to be about 700 lb. per lineal inch of joint. For belts that are to be used in damp places brass, instead of steel, wire is recommended.

CHAPTER VII

MISCELLANEOUS NOTES ON BELTING

U. S. Navy Department Specifications for Belting. The specifications for leather belting issued by the Navy Department call for belts to be made from No. 1 native packer steer hides or their equal. They are required to be tanned with white or chestnut oak by the slow process for six or eight months; chemical processes are not permitted. The leather is to be thoroughly curried by hand and must not be stuffed or loaded for artificial weight. The leather must not crack open on the grain side when doubled strongly by hand with the grain side out. All belting must be cut from the central part of the hide, not more than 15 in. from the backbone or more than 48 in. from the tail toward the shoulder. Belts 8 in. wide and over must be cut to include the backbone. All leather must be stretched 6 in. in the lengthwise direction and should not exceed 54 in. after stretching. Centers and sides are to be stretched 6 in. separately. That is, all sides from which widths of less than 8 in. are cut must be stretched after the backbone section has been removed. Center sections are to be stretched in the same size for which they are to be used.

For single belts up to 6 in. wide the laps must not exceed 6 in. nor be less than $3\frac{1}{2}$ in. long. For single belts wider than 6 in., the laps should not be more than 1 in. longer than the width of the belt. For double belts laps must not be more than $5\frac{1}{2}$ in. nor less than $3\frac{1}{2}$ in. long. Filling straps are not permitted. All laps must be held securely at every point with the best quality of belt cement, and when pulled apart must not show any resinous, oily, vitreous or watery condition. The belts should be again stretched after manufacture.

Belting is to weigh as follows: Single belts, all sizes, 16 oz. per square foot; double belts: 1 to 2 in. wide, 26 oz. per square foot; $2\frac{1}{2}$ to 4 in. wide, 28 oz. per square foot; $4\frac{1}{2}$ to $5\frac{1}{2}$ in. wide, 30 oz. per square foot; 6 in. wide and over, 32 oz. per square foot.

Rawhide Lacing. Only hand-cut green slaughter-house hides are to be used for lacing. They are to be cut in the following sizes: $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$, $\frac{1}{2}$, $\frac{5}{8}$ and $\frac{3}{4}$ in. They must be cut lengthwise from the hide and have a tensile strength of not less than

Width, in.	1/4	5/16	3/8	7/16	1/2	5/8	3/4
Tensile strength, lb. per sq. in.	95	125	155	165	180	205	230

Weight and Thickness of Belting. Samuel Webber in the *American Machinist*, May 11, 1909, published notes on leather belting, from which the following figures of the weight and thickness of the different classes of belting are abstracted:

	Single.	Light Double.	Medium Double.	Standard Double.	Three-ply.
Weight, lb. per sq. ft.	16	24	28	33	45
Approximate thickness, in.	1/8	1/4	5/16	1/3	9/16
Actual thickness, in.	0.16	0.24	0.28	0.33	0.45

Good oak-tanned leather belting from the back of the hide is stated to weigh almost exactly 1 oz. avoirdupois for each $\frac{1}{100}$ in. of thickness in a piece of leather 1 ft. square.

Belt Dressing. The use of a belt dressing is to increase the coefficient of friction of a belt and thus enable it to pull a heavier load with a given tension. The dressing should be applied only when the belt slips during working hours and it is impracticable to tighten it before the shop shuts down for the day or the noon recess. Belts which are regularly inspected and whose tensions are maintained according to the rules given in the chapters on horsepower of belts and belt maintenance will seldom slip, and need no dressing except the regular cleaning and greasing every five or six

months. If, however, the belt should slip during working hours as a result of neglect or of being subjected to a heavier load than that for which it was intended, a dressing should be applied sparingly to enable it to continue to pull its load until the machine can be shut down and the belt tightened.

The Tabor Manufacturing Company, Philadelphia, in the instructions issued with the belt benches described in Chapter V, recommends the cleaning of belts at intervals of six months and the application of a grease that will give to the surface a clean sticky pulling surface, similar to that which it has when new due to the oils and grease that it contains. The grease in the new belt becomes exhausted after the belt has been in service for a certain length of time, as the result of heat, friction and atmospheric action, and the application of additional grease is necessary if the belt is to be maintained in a soft and pliable condition. The grease recommended is the same as that put in the new belt by the manufacturers and consists of

Edible beef tallow	2 lb.
Cod liver oil	1 lb.

The tallow is melted and allowed to cool slightly, after which the cod liver oil is added and the mixture stirred in one direction until cold. This grease can be purchased mixed ready for use from Alexander Bros., Philadelphia. It should be applied in light coats by means of a camel's hair brush, and only in such quantities as the belt will easily absorb.

For use on belts which slip in service the same company recommends the "Plomo" dressing, made by the Plomo Specialty Co., Cleveland. A few drops of this, applied to the belt will enable it to pull its load for several hours, until the opportunity presents itself for tightening the belt. The dressing, however, should not be regarded as a preservative, and its use should be limited to the purpose described above. It should be used sparingly, as a too liberal applica-

tion will harm the belt by causing it to slip and thereby injuring the surface.

Many mechanics, when the belt on their machine slips, apply rosin to the surface of the belt. This practice cannot be too severely condemned. The rosin may for a time remedy the slipping, but it will be at the cost of a ruined belt. The use of rosin on any belt should be absolutely prohibited in every factory, and after such prohibition has been made known to the employees, any person guilty of using it should forthwith be dismissed.

Cleaning of Belts. Every precaution should be taken to prevent belts in service becoming saturated with lubricating oil, or, in fact, from coming in contact at all with any kind of oil. If, however, a belt does become so saturated it may be cleaned by first scraping it and then packing it in sawdust or other absorbent for several days. A quicker method is to pass the belt between rolls, under pressure, thereby squeezing out the oil which then may be absorbed by powdered chalk. The surface dirt may be removed from belts by wiping them with a cloth moistened with kerosene. This should be done in every case before applying belt grease.

Utilization of Old Belting. Old belts which are considered worthless often may contain sections which are capable of further use. These sections may be utilized by first running them through rolls to squeeze the oil out of them, which is absorbed with powdered chalk. They then should be scraped clean and the pieces cemented together as described in Chapter VI. The adoption of this practice will result in the saving of many feet of belting that otherwise would be scrapped, and will materially cut down the cost of new belting.

Direction in which Belts Should Run. The thin edge of all splices and laps on the side of the belt next to the pulley should point away from the pulley which it is approaching. The hair side of the belt should be next to the pulley.

Causes of the Destruction of Belts. In those shops where the belting problem has not been given the attention that its importance deserves, few belts become useless as the result of legitimate service and wear, but fail due to misuse, accidents, and lack of care. One of the commonest causes of belt failures is the use of belts that are too wide for the pulleys. If the pulleys are out of line only slightly, a wide belt will run over one or the other edge of the pulleys, and come in contact with any fixed object nearby, such as the belt shifter, the edge of the next step of a cone pulley, a post or column close to the belt run, or any other projection.

Careless splicing also is a frequent cause of ruined belts, particularly where rawhide lacing is used for the splicing. If the ends of the belt are not cut square, or if the tension put in the lacing is not equal at the two sides of the belt, the belt will have a tendency to run crooked, thus leading to the troubles enumerated in the preceding paragraph.

Another prolific cause of belt trouble, particularly in those shops where the care of the belts on the different machines is entrusted to the man in charge of the machine, is the practice of running the belt on " cross cones," that is on two steps of the cone pulleys that are not in line. This is an easy way for the so-called mechanic who is too lazy to take care of his belts to make a loose belt tight or a tight belt loose, but it is fatal to the belt. The edge of the belt is compelled to rub against the edge of the cone pulley steps and curls. The layers of the belt are separated, and the belt is stretched along one edge. A very short service under such conditions will ruin a belt beyond any hope of repair.

Belt shifters which do not throw the belt entirely off or on the loose pulley are also frequent contributors to the destruction of belts. A belt running partly on and partly off a loose pulley will stretch unevenly and soon be spoiled.

Inattention to the splices in the belts when they begin to give evidences of weakness and to separate will also help to

increase the belting cost. Regular inspection of the belts will detect failures in the splices and enable them to be repaired before they have gone far enough to do serious damage.

The too liberal use of belt dressings whose object is to prevent the belt from slipping will shorten the life of the belt, as will the failure to clean and grease it at regular intervals. A belt allowed to become dry, hard and stiff will soon begin to crack and break. The application of rosin will ruin a belt more quickly than almost any other sort of abuse. Lubricating oil is injurious to belting and should be guarded against. Where a belt has been allowed to come in contact with oil, it should be cleaned at the earliest opportunity. Moisture, steam and water are harmful to leather belts, and for service in damp locations the use of rubber belts is recommended. Excessive temperatures also will damage belts and the usual belt will soon deteriorate if subjected to more than 130 degrees Fahr. "Duxbak" belting, however, is advertised to withstand any amount of moisture and temperatures up to 200 degrees.

Finally, it will pay to use only the best grade of belting, and to use belts of ample size and weight. Oak-tanned and fulled leather belts will as a rule give the best service, and should be used double on pulleys of 6 in. diameter and over. Belts of good quality, regularly inspected, well cared for, and run under the tensions recommended in the foregoing pages should last from ten to fifteen years. Belts which are neglected or abused will not last over two or three years and may give out in even less time.

Effect of Humidity on Leather Belts. Prof. W. W. Bird, of the Worcester Polytechnic Institute, and F. W. Roy, reported to the American Society of Mechanical Engineers[1] the results of tests made upon a 4-in. leather belt to determine the effect of humidity on the tension of the belts. In

[1] Transactions, American Society of Mechanical Engineers, 1915.

the tests the horsepower transmitted was maintained at a constant figure, while the humidity was varied. Tests were made at varying center distances. The tests showed that as the humidity decreased the sum of the tensions in the belt increased. The results of the tests, as taken from curves presented in the paper are given in the following table:

EFFECT OF HUMIDITY ON LEATHER BELTS

Relative humidity.	Center Distances.			
	9 ft. 6 in.	9 ft. 6½ in.	9 ft. 7 in.	9 ft. 7½ in.
	Sum of the Tensions, lb.			
90	95	210	325	445
55	125	260	400	550
20	150	310	465	620

It was shown that increasing the temperature as well as the humidity tended to increase the length of the belt and so to decrease the tension. The principal conclusions drawn from the tests were as follows: 1. If a belt be set up at a moderate relative humidity, the tensions will not be excessively lower at lower relative humidities, nor will there be any great danger of the belt slipping at higher relative humidities unless there are excessive temperature changes. 2. If a belt be set up at any relative humidity with a spring or gravity tightener, a load 50 per cent greater than the standard can be transmitted at either high or low humidity without danger of slipping, stretching the belt or producing excessive pressure on the bearings.

To Find the Arc of Contact of the Belt on the Pulley. The angle of contact of the belt on the smaller of a pair of pulleys is the difference between 180 degrees and twice the angle between the diameter perpendicular to the line joining the centers of the pulleys and the radius drawn to the point of tangency of the belt on the pulley. In Fig. 18 it is the angle $cbf = 180 - 2(\text{angle } abc)$. The angle abc is similar to

the angle *ebd*, which is the angle whose sine is $de \div db$. It can readily be seen from the figure that *de* is the difference between the radius of the larger pulley and that of the smaller one, and that *db* is the center line distance of the pulleys themselves. If then we denote the radius of the larger pulley by R and that of the smaller one by r, and the distance between their centers by L, all being in the same unit, as

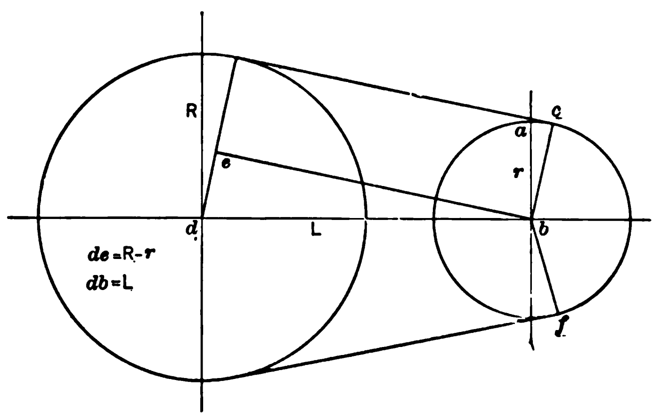


FIG. 18.—METHOD OF DETERMINING ARC OF CONTACT OF BELT ON PULLEY.

inches or feet, the arc of contact, in degrees, of the belt on the smaller of a pair of pulleys will be

$$180-2\left(\text{angle whose sine is } \frac{R-r}{L}\right).$$

The angle of contact on the larger pulley will be

$$180+2\left(\text{angle whose sine is } \frac{R-r}{L}\right).$$

The length of belt in contact with the larger pulley is found by multiplying the angle whose sine is $\frac{R-r}{L}$ as above by

0.0349, adding π to the product and multiplying the sum by the radius of the pulley. Or

$$\text{Length} = \text{radius} \times \left(\pi + 0.0349 \sin^{-1} \frac{R-r}{L}\right).$$

If the length of belt in contact with the smaller pulley is desired, the formula will be

$$\text{Length} = \text{radius} \times \left(\pi - 0.0349 \sin^{-1} \frac{R-r}{L}\right).$$

Length of Belts. Whenever possible the length of belt required for a pair of pulleys should be determined by actual measurement by means of a steel tape passed around the pulleys. Where this is impracticable, the length of an open belt can be determined by calculation, using one of the three formulæ below published by Carl G. Barth in the *American Machinist*, March 12, 1903:

$$L = \frac{(D+d)\pi}{2} + Cx + 2C. \quad \ldots \ldots \quad (1)$$

$$L = \frac{(D+d)\pi}{2} + Cx\frac{12}{12-x} + 2C. \quad \ldots \quad (2)$$

$$L = \frac{(D+d)\pi}{2} + Cx\frac{60-13x}{60-18x} + 2C. \quad \ldots \quad (3)$$

L is the length of the belt, D the diameter of the larger pulley and d the diameter of the smaller pulley, and C is the center line distance of the pulleys, all in the same unit, inches or feet as the case may be, $\pi = 3.1416$, and $x = \left(\frac{D-d}{2C}\right)^2$. The accuracy of the formulæ is as follows, testing them by the limiting case where $d = 0$ and $C = \frac{D}{2}$, giving $L = D\pi$: Formula (1) gives a result somewhat over 2 per cent short; formula (2) gives a result about 1 per cent short; formula (3) gives a result under 0.4 per cent short.

Crossed Belts. The length of a crossed belt may be obtained exactly by the formula

$$L=2\left\{\sqrt{C^2-\left(\frac{D+d}{2}\right)^2}+D\left(\pi-\cos^{-1}\frac{D+d}{2C}\right)\right\}.$$

The notation is the same as in the three formulæ above for open belts.

Length of a Roll of Belting. The length of a belt, in feet, when closely rolled is approximately equal to the sum of the diameter of the roll and of the eye of the roll in inches, multiplied by 0.1309 times the number of turns.

Idle Belts. It is advisable when belts are out of service for any length of time to throw them from the pulleys. They should be suspended from hooks, however, and not allowed to rest upon the shafting. This latter practice, which is altogether too common, is not only destructive of the belts, but is frequently the cause of accidents. The arrangement shown in Fig. 19 should be used where belts are removed from the pulleys. The hook, of $\frac{3}{8}$ in. round iron, is suspended from the ceiling by a staple. It should be long enough to reach almost to the rim of the pulley, and should swing the belt clear of the shaft.

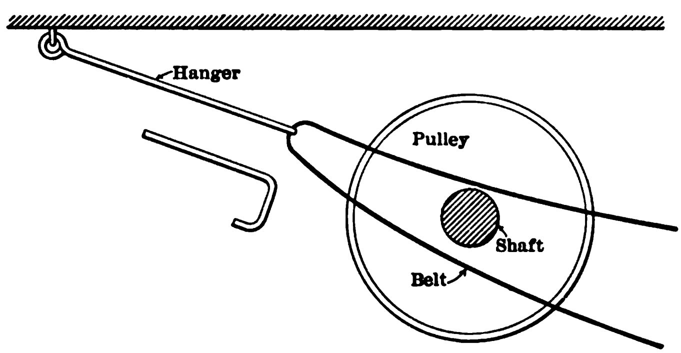


FIG. 19.—HANGER FOR IDLE BELT.

CHAPTER VIII

PULLEYS

THE life of and service rendered by belting is affected in no little degree by the condition and arrangement of the pulleys on which it runs. The notes on pulleys which follow cover those features that have a direct influence on the belt problem. In general it is advisable to use as large pulleys as possible, providing that they are not so large as to give a belt velocity in excess of 4700 ft. per minute. Small pulleys running at high speed, however, will cost less when equipping the mill than will large pulleys running at low speeds.

Width of Pulleys. The face of the pulley should always be wider than the belt that is to be used on it. Carl G. Barth (*American Machinist*, February 11, 1915) gives the following formula for the relation of the width of pulley to that of the belt, in which F and B are the widths of the pulley and belt respectively, both in inches:

$$F = 1\tfrac{3}{16}B + \tfrac{3}{8}.$$

The limits of design may make it impractical to use pulleys as wide as those called for by the equation just given, in which case the following may be substituted:

$$F = 1\tfrac{3}{32}B + \tfrac{3}{16}.$$

Values of F by both formulæ for various values of B are given in Table VI on page 110.

Crowning of Pulleys. A belt on a pulley tends to climb to the largest diameter of the pulley. Advantage may be taken of this tendency to compel the belt to run in the center of the pulley face by making the pulley of slightly larger diameter at the center than it is at the edges, or "crowning" it. The amount of crown that should be given to a pulley

varies according to different authorities, as follows, F being the width of the pulley and H the height of the crown, both in inches:

Morin. $H = 0.1F$
Molesworth. . . . . $H = \frac{1}{24}F$
Barth. . . . . $H = 0.03125\sqrt[3]{F^2}$

Scott A. Smith says that the crown should not be over $\frac{1}{8}$ in. for a pulley of 24 in. face. The chart, Fig. 20, gives the

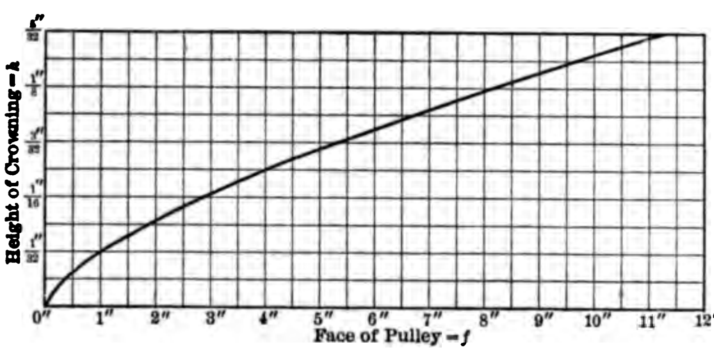


FIG. 20.—RELATION OF HEIGHT OF CROWN TO WIDTH OF PULLEY.

height of crown for pulleys up to 12 in. face according to Barth's formula.

Arrangement of Pulleys. If possible machinery driven from overhead shafting should be so arranged that the belt will make an angle from the vertical of not less than 45 degrees. Never, unless it is absolutely impossible to avoid it, should the driving and driven pulleys be in the same vertical line. Care should be taken to arrange the pulleys in line with each other, the tops of the crowns being aligned. Otherwise the belt will tend to stretch unevenly and to run crooked. Machinery should be so placed that the tight strand of the belt will be the bottom strand. If, however, the machines are so placed that the pull of the belts at alternate line shaft pulleys is in opposite directions the wear and strain on bear-

ings will be reduced. This arrangement of machinery will make one-half of the belts drive with the top strand tight.

The distance between the shafts has considerable influence on the life of belts, and the distances should as a rule be made as great as possible up to reasonable limits. For narrow belts on small pulleys, a good average figure is 15 ft.; for larger belts the center line distance should be from 20 to 25 ft., while for main belt drives using wide and heavy belts the pulleys

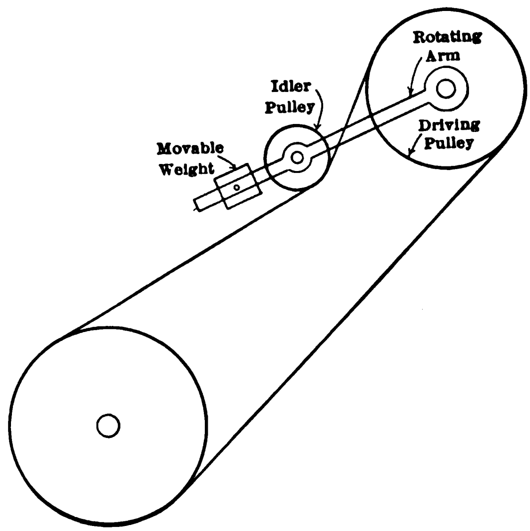


FIG. 21.—IDLER PULLEY ARRANGED AS A GRAVITY TIGHTENER.

should be from 25 to 30 ft. apart. If the distance is made too great the belt will have a tendency to flap and injure itself.

Idler Pulleys. Belts which are run on pulleys located close together are liable to stretch rapidly and thus lose the tension necessary to the transmission of the desired amount of power. For such cases, and also for belts which are required to run for long periods without stopping, an idler pulley which can be moved against the belt to take up the slack and thus increase the tension, will be found desirable. The idler should be located about one-quarter of the distance

from the driving pulley and should bear against the slack side of the belt. Its effect is not only to increase the tension, but also the arc of contact. The idler may be so constructed that it can be moved against the belt at will, or it may act automatically by means of a spring or gravity. Such an arrangement is shown in Fig. 21.

Quarter-twist Belts. If two shafts are at right angles to each other, and in different planes, for instance an engine shaft near the floor and a line shaft on the ceiling, and one is to be driven from the other, a belt with a quarter twist in it may be used, providing that the distance between the shafts is not too short. The belt must run squarely on to each pulley and the point at which the belt leaves each pulley must lie in the straight line formed by the intersection of the planes passing through the center line of the face of each pulley and perpendicular to their axes. This is shown in the illustration Fig. 22. The driving pulley is *AD* and the belt runs in the direction shown by the arrows. The point *C* at the end of the horizontal diameter of the driven pulley at which the belt leaves it is set so that it is vertically above the point *A* at the end of the horizontal diameter of the driving pulley where the belt leaves that pulley. Should the shafts in question be respectively a horizontal and a vertical one the relative arrangement of the pulleys would be the same as shown in the illustration. In fact, it makes no difference what position the shafts take so long as the leaving points of the belts on the pulleys are in the straight line determined by the intersection of the two planes as described. The quarter-twist belt can be used only with narrow belts—say up to 6 in. wide—and where the distance between the shafts is considerable. For heavy belts and short center line distances, guide pulleys or mule pulley stands should be substituted as shown below. A quarter-twist belt will drive in one direction only.

Where the quarter-twist belt drives through a floor the

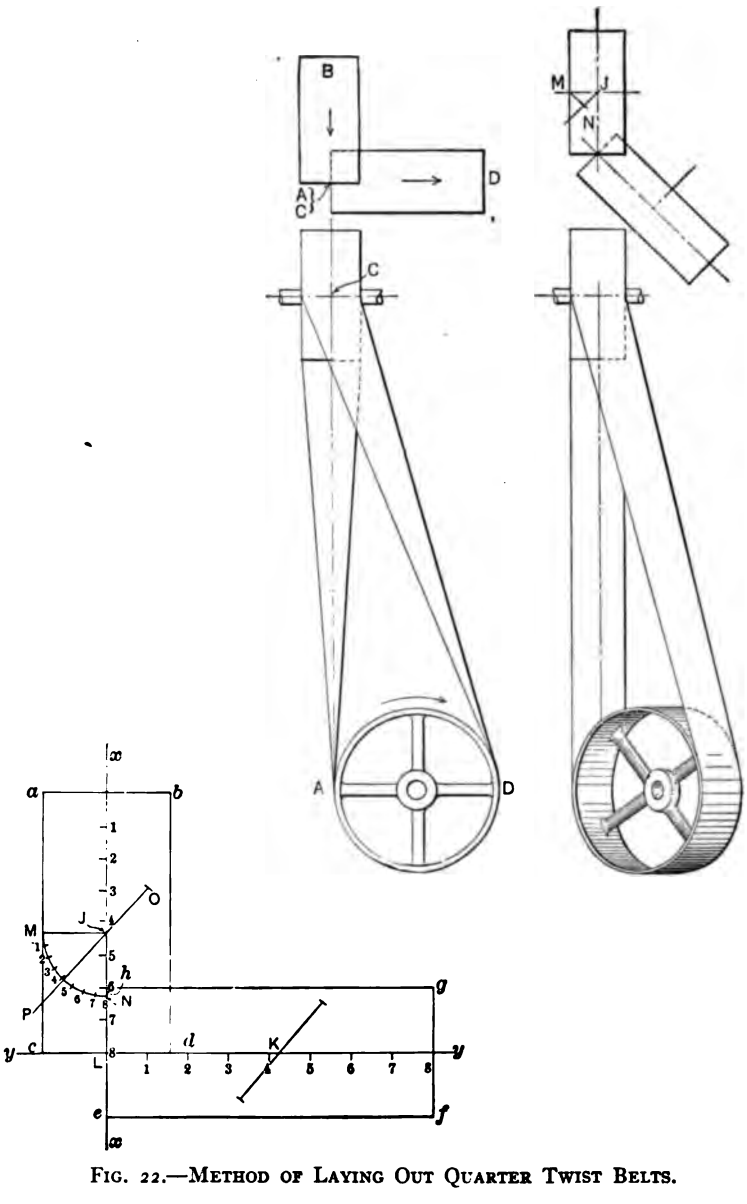


FIG. 22.—METHOD OF LAYING OUT QUARTER TWIST BELTS.

holes for the belt must be laid out with some care. A method of laying out these holes is described in *Machinery*, September, 1912, by M. H. Ball, and is reproduced below and in Fig. 22. From the end of the horizontal diameter at which the belt leaves the upper pulley a plumb line is dropped to the floor, and the point where it touches the floor is marked. This is point L in the illustration. Through L are drawn on the floor the lines XX and YY respectively perpendicular to the lines of shafting on which the upper and lower pulleys are carried. The outlines of the pulleys in plan are then traced on the floor, these being $ABXDLC$ and $EFGHL$ respectively for the upper and lower pulleys, each being symmetrical about the center lines XX and YY. These center lines, which are the projections of the horizontal diameters of the pulleys, are next divided into eight equal parts, the divisions being numbered as shown. The numbers should start at the end of the diameters toward which the pulley is rotating, that is at the end of the diameter at which the belt runs on the pulley. Next divide the distance in inches between the center lines of the shafts by the number of divisions laid out on the pulley, and the distance from the center line of the upper shaft to the floor by the quotient so obtained. The latter figure represents the number of spaces on the pulley that the belt at the floor will be away from the end of the diameter at which it runs on to the pulleys, that is, the points J and K in the illustration. From J as a center the arc MN is struck with the line JM, perpendicular to XX, as a radius. The arc is divided into eight equal parts and numbered from M to N. The same number of spaces are taken on the arc as were taken on the lines XX and YY, and the line OP is drawn through the point J and the point on the arc so found, PJ being made equal to JO. This line is the position of the center line of one strand of the belt at the floor, while a line parallel to OP through K, and of the same length is the position of the center line of the other strand.

Quarter-twist belts may be used to drive shafts which make angles with one another of other than 90 degrees, if the rule is observed that the points at which the belt leaves the pulleys are in the straight line determined as explained previously. The method of laying out holes in the floor is the same as that used for shafts at right angles, except that the arc *MN* on the pulley outline extends only to a line drawn from the point *J* parallel to the face of the lower pulley. This is the

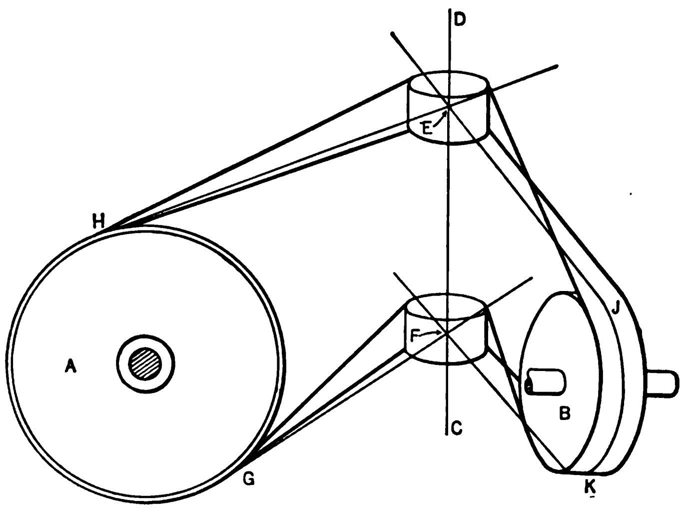


Fig. 23.—Method of Locating Guide Pulleys.

line *JR* in the sketch showing the angular arrangement of pulleys in Fig. 22.

Guide Pulleys. Where two shafts that are not parallel and that do not intersect are to be connected by belting, and the conditions are such as to make a quarter-twist belt inadvisable, guide pulleys are used. The function of the guide pulley is to alter the direction of the belt and cause it to run squarely onto the receiving pulley. The location of the guide pulley with the reference to the main pulleys is determined by the intersection of the planes passing through the center of the

face of the main pulleys and perpendicular to their axes. Thus in Fig. 23, let *A* and *B* represent the main pulleys, and *xx* and *yy* the planes passed through the middle points of their respective faces. These intersect in the line *CD*.

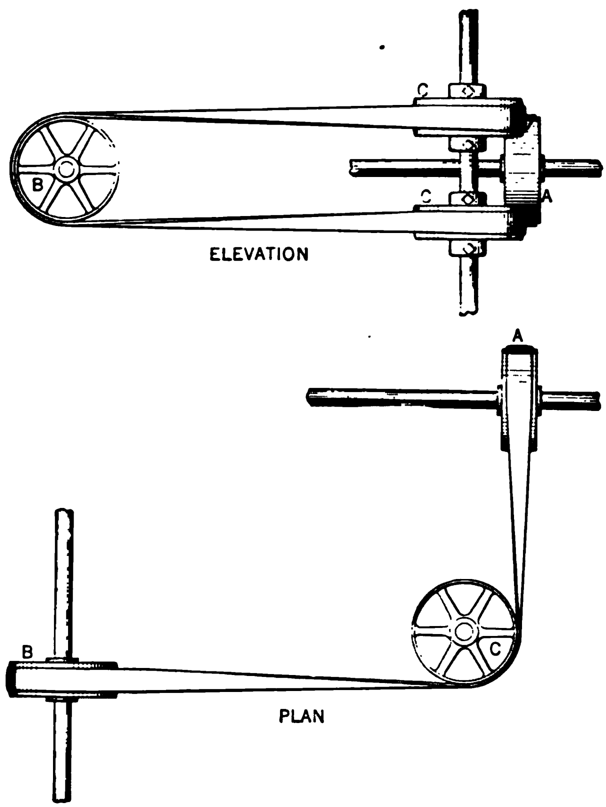


Fig. 24.—Guide Pulleys for Shafts at or Near Right Angles.
C C, loose pulleys. The belt will run in either direction.

Tangents from any two points as *E* and *F* on this line to the points *H* and *J*, and *G* and *K* respectively on the pulleys *A* and *B* will determine proper paths for the belt. The guide pulleys should be located at the intersection of the tangents *HE* and *EJ* and *FG* and *FK* respectively and the plane

through the center of the face of each pulley should coincide with the planes formed by these tangents. The illustrations Figs. 24 to 27 show various arrangements of guide pulleys. Others will suggest themselves according to the needs of each particular case. It is only necessary to observe the precaution that the guide pulleys be so placed that they will receive the belt squarely from the main pulley and deliver it squarely to the next one. If this be done, the pulleys may be placed in practically any location and at practically any angle.

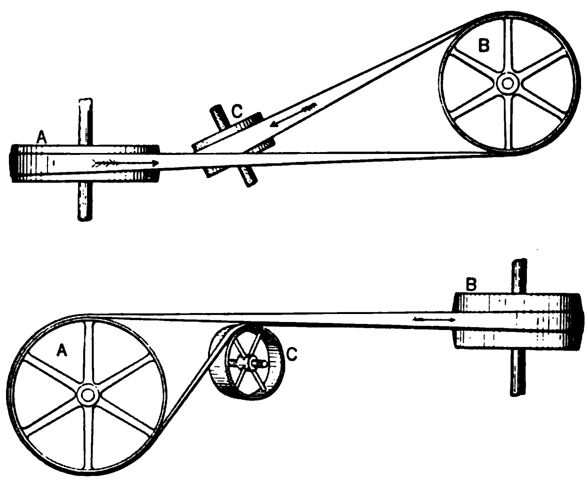


FIG. 25.—SINGLE GUIDE PULLEY.

A common arrangement of guide pulleys, known as a mule pulley stand, is shown in Figs. 28 and 29. The method followed in locating the pulleys and determining the angles at which they are to be placed is as follows: The pulleys which are to be connected by a belt are A and B, and they have the relative locations as shown in the various views. The center line XX in the elevation is prolonged, and at a distance L (see plan) from the line ZZ through the middle plane of the face of the pulley A, the line $Z'Z'$ is drawn. Through the point p, the line YY is drawn parallel to XX,

and at a distance below it equal to the vertical distance between the shafts of the two pulleys. With p as a center

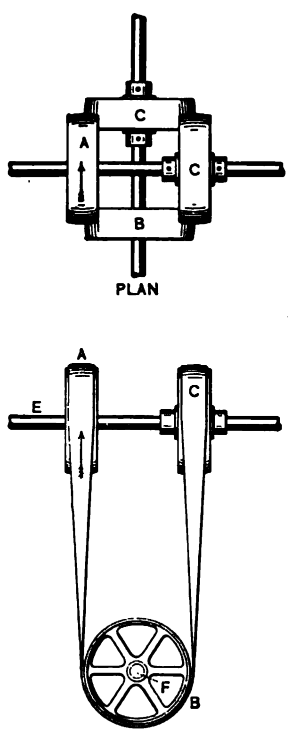


Fig. 26.
C C, Loose Pulleys.

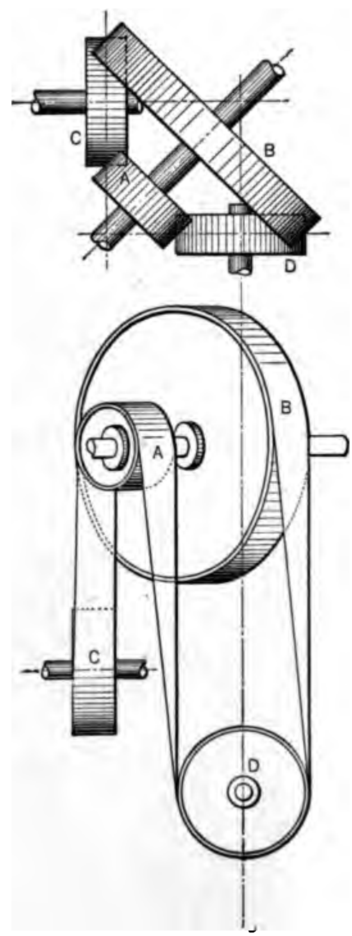


Fig. 27.
A, Loose Pulley.

Guide Pulleys.

the circle A' is drawn with a radius equal to the radius of the pulley A. The circles B and A' are then joined by the

Page 82 y 83

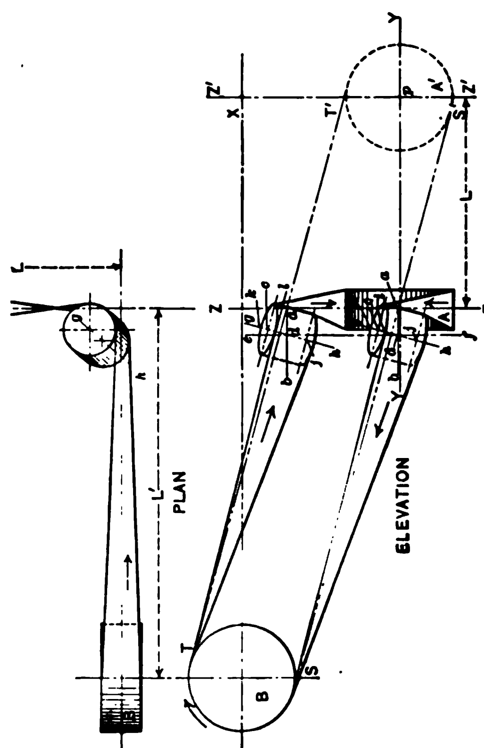


FIG. 28.
LAYING OUT A MULE PULLEY STAND.

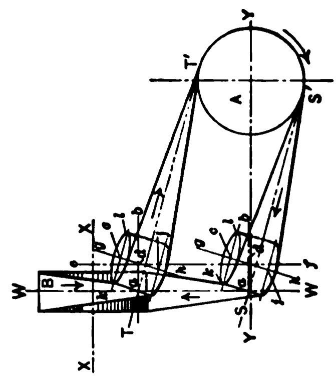


FIG. 29.

tangents TT' and SS', which represent the inclination of the belt with respect to the pulleys at all points along its length. At the point a where these tangents intersect the line ZZ the lines ab are drawn parallel to the line XX. The diameter of the mule pulleys having been decided, the point d is set off on ab at a distance from ZZ equal to half the diameter of the mule pulley. Through d is drawn the line ef parallel to ZZ. This represents the center of the mule stand. The lines gh through the point d, perpendicular to the tangent TT' or SS' will be the axis of the mule pulley. The points c and j are laid off on either side of the point d on the line gh, dc and dj being made equal to half the width of the mule pulley. The bearings of the mule pulley will lie on the line gh outside of the points c and j, and their position with reference to the center lines of the main pulleys can be ascertained by scaling the drawing or by calculation, as preferred. If the drawing is laid out to a large enough scale, scaling will be accurate enough for all practical purposes. The axes of the mule pulleys will necessarily be inclined from the vertical, not only as regards the plane ZZ through the middle of the face of the pulley A, as shown in the elevation Fig. 28, but also as regards the plane WW through the middle of the face of pulley B as shown in Fig. 29, and this angle will be the same in the two views. Lines through the points c and j parallel to the tangents TT' and SS' will be the major axes of the ellipses representing the upper and lower sides of the mule pulleys in the two views in Figs. 28 and 29. The position of the center of the mule stand and of the mule pulleys in the end view of the drive, Fig. 29, is determined in exactly the same manner as the similar points in the elevation in Fig. 28, and no further explanation is necessary. The various points in Fig. 29 are designated by the same letters as the corresponding points in Fig. 28, and the same explanation will apply.

If it be borne in mind that the governing principle of

mule pulley drives, and in fact of all drives where the belt joins two pulleys that are not in the same plane, is that the belt must be delivered squarely on the pulleys, but that it can leave them at any angle within reasonable limits, it is evident that the mule pulleys can be placed in any location desired. If local conditions make it advisable, the mule pulleys can be so placed as to require the belt to leave either or both of the main driving pulleys in a horizontal position. The method of laying out the drive is the same in any event. The only variation will be in the relative length of the dimensions L and L'. The sum of these is taken and the position of the circle A' is then determined as before. The angle of the pulleys will be the same, as will the length of the belts.

The fact that it is possible to vary the position of the mule pulleys without affecting the working of the drive makes it possible to place both mule pulleys on the same shaft. Referring to Fig. 28, if the belt travels in the direction of the arrows, it is evident that if the lower mule pulley be moved in the direction that the belt travels, the belt will still be delivered squarely to pulley B. It will, of course, leave pulley A at an angle, but this will have no effect on it if the mule pulley is maintained at the same angle as before, since the belt will still be received squarely by the mule pulley. If the distance between the axes of the mule pulley shafts is not excessive, the lower mule pulley may be moved until the two shafts are in line, as regards Fig. 28. They will, however, still be out of line as regards Fig. 29. By the same process of reasoning as before, however, it is evident that the upper mule pulley can be similarly moved in the direction of the belt travel until its axis is in line with that of the lower pulley in Fig. 29. The axes of the two mule pulleys then will coincide, and in their new locations they can be mounted on the same shaft.

The above considerations apply to drives where the main

pulleys are of the same diameter. If they are unequal, the shafts of the mule pulleys will lie at different angles and the pulleys must, perforce, be mounted on separate shafts. The location of the shafts is determined by the same method as described above, however, the only difference being that the axis of the upper mule pulley is determined by the perpendicular to the tangent TT', Fig. 28, while that of the lower one is determined by the perpendicular to the tangent SS'. These tangents in this case will not be parallel. Otherwise the

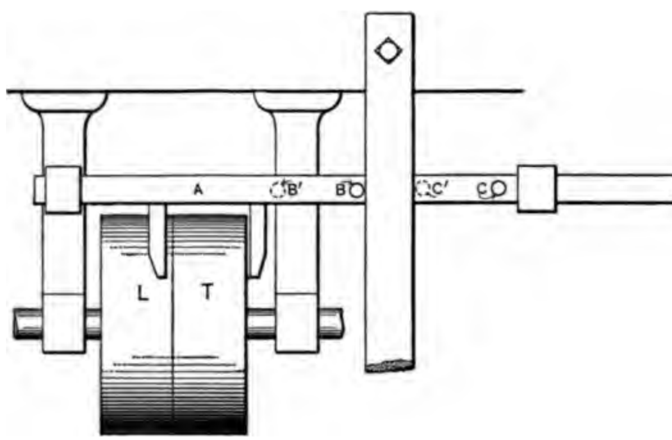


FIG. 30.—BELT SHIFTER.

construction is the same as for drives with main pulleys of equal diameter.

Belt Shifters. The fault of the ordinary belt shifter consisting of a fork pushed to one side or the other, is that frequently the weight of the pole operating the shifter will, when it is at one side or the other of the vertical, tend to cause it to resume its vertical position and so position the belt that it is partly on and partly off the loose pulley. This can be obviated by the arrangement shown in Fig. 30. The pins on the fork bar *A* are spaced apart a distance twice that through which the belt is shifted. Whatever the position

of the belt on either the tight or loose pulley, the shifter pole hangs vertically and bears against one or the other of the pins. Thus, in the illustration the pole resting against the pin B is in position to move the belt on to the pulley L. In doing this the pin B will move to B'; the pin C will take the position C', and the shifter pole will rest against it as soon as the pole is released after moving the belt on to the pulley L, thereby being in position to move the belt back to pulley T.

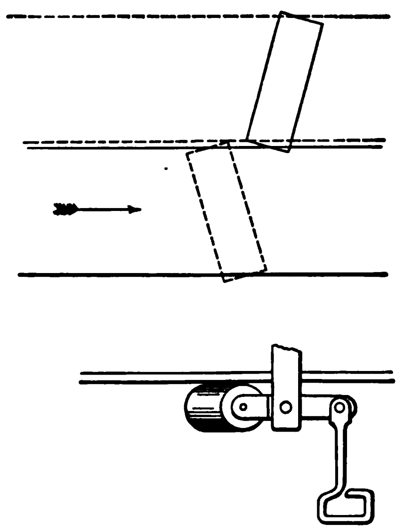


FIG. 31.—ROLLERS FOR SHIFTING HEAVY BELTS.

The belt fork has no tendency to move from the position in which it is left by the pole, and if the pulleys are crowned, the belt also will remain in the position where it is left.

For shifting wide and heavy belts a pair of rollers making an angle of about 75 degrees with the center line of the belt are advisable. These rollers may be arranged so that they can be pressed against the belt, whereupon the belt will be forced over on to the other pulley. This arrangement is shown in Fig. 31.

BELTING TABLES

METHOD OF USING THE BELTING TABLES

1. Read carefully the explanation in Chapter III.

2. Determine, from Table II, the arc of contact of the belt on the smaller pulley. Subtract the diameter of the smaller pulley from that of the larger. In the column under "Difference in Diameter of Pulleys" corresponding to the difference so obtained, read the arc of contact opposite the proper center to center distance of pulleys.

3. Ascertain from Table III the velocity of the belt in feet per minute.

4. In that portion of Table I pertaining to machine or countershaft belts, as the case may be, and in the "arc of contact" section most nearly corresponding to the arc of contact as found above from Table II, read the horsepower that will be transmitted per square inch of belt section at the velocity as found from Table III. Multiply the figure so found by the area of the belt as found in Table IV, to ascertain the total horsepower that the belt will transmit at the given speed.

5. Opposite the horsepower as found in Table I, read the maximum and minimum tensions per square inch of belt section at which the belt should be operated. Multiply these tensions by the area of the belt to ascertain the total maximum and minimum tensions for the belt. These are the figures that are to be entered on the belt-fixer's order, Fig. 12, page 53.

6. If the velocity of the belt as found from Table III is intermediate to the velocities given in Table I, subtract the

horsepower opposite the next lower velocity in Table I from the horsepower opposite the next higher velocity. Multiply the difference by the quotient obtained by dividing the difference between the actual belt velocity as found from Table III and the next lower velocity in Table I, by the difference between the next lower and next higher velocities in Table I. Add the product to the horsepower opposite the next lower velocity than the actual. The sum is the total horsepower transmitted by a belt of 1 sq. in. of cross-section at the actual velocity. Multiply this sum by the cross-sectional area of the belt to find the total horsepower.

7. To find the maximum and minimum tensions for belts at velocities intermediate to those given in Table I, multiply the difference in tensions at the next higher and next lower velocities to the actual velocity as given in Table I, by the quotient found as described in the preceding paragraph. If the tension as given in Table I at the lower velocity is higher than the tension given for the next higher velocity, *subtract* the product so found from the tension given in the table for the next lower velocity. If the tension given in the table is higher at the higher than at the lower velocity, *add* the product to the tension given opposite the next lower velocity. In either case multiply the sum or difference so obtained by the area of the belt to ascertain the total maximum tension (t_m) or total minimum tension (t_o), as the case may be.

8. To ascertain the size of belt that will be required for a given horsepower, determine the horsepower that a belt of 1 sq. in. of cross-section will transmit under the conditions of speed and arc of contact that exist, by the method given in the first part of paragraph 4, or in paragraph 6. Divide the figure so obtained into the horsepower to be transmitted. This will be the cross-scetional area of the belt. From Table VI find the maximum width of belt that can be used on the pulleys given, and then opposite this width

in Table IV find the figure that most nearly corresponds to the cross-sectional area. At the head of the column read the thickness of the belt.

9. To ascertain the amount that is to be taken out of a belt to restore it to its maximum tension, subtract the tension in the belt before tightening from the maximum tension. Find the figure corresponding to this difference at the bottom of the chart Fig. 32, and run vertically upward on the chart until the line corresponding to the cross-section of the belt is encountered. At the left of the chart, opposite the intersection of the tension and the area so found, read the stretch per 18 in. of length of belt. Multiply this figure by the quotient of the length of the belt in inches divided by 18, or of its length in feet divided by 1.5. The result will be the amount that must be taken out of the belt to restore it to its maximum initial tension.

Table I

Horsepower and Tensions of Belts

Velocity of Belt. Ft. per min.	Machine Belts.						Countershaft Belts.					
	Horsepower.	Initial tension, lb. t_m.	Minimum Tension, lb. t_0.	Tension on Tight Side, lb. t_1.	Tension on Slack Side, lb. t_2.	Effective Pull, lb. t_1-t_2.	Horsepower.	Initial Tension, lb. t_m.	Minimum Tension, lb. t_0.	Tension of Tight Side, lb. t_1.	Tension on Slack Side, lb. t_2.	Effective Pull, lb., t_1-t_2.
180 Degrees Arc of Contact.												
250	1.05	183.75	127.5	206.0	67.9	138.1	0.70	141.0	85.0	137.3	45.3	92.0
500	2.27	180.25	124.0	209.8	60.3	149.5	1.51	139.0	82.5	139.8	40.4	99.4
750	3.53	179.0	121.75	211.8	56.5	155.3	2.34	138.0	81.25	141.1	37.9	103.2
1060	4.78	178.5	121.0	212.5	54.9	157.6	3.15	137.75	81.0	141.3	37.4	103.9
1200	5.77	178.0	120.5	212.9	54.3	158.6	3.78	137.75	81.0	141.3	37.3	104.0
1400	6.72	178.0	120.5	212.8	54.4	158.4	4.38	138.0	81.25	141.1	37.8	103.3
1600	7.64	178.25	121.0	212.5	54.9	157.6	4.96	138.25	81.75	140.7	38.7	102.0
1800	8.53	178.5	121.5	212.1	55.8	156.3	5.47	138.75	82.25	140.1	39.8	100.3
2000	9.36	179.25	122.0	211.5	57.0	154.5	5.95	139.0	83.0	139.4	41.3	98.1
2500	11.20	181.0	124.5	209.3	61.5	147.8	6.88	141.25	85.5	137.0	46.1	90.9
3000	12.62	183.5	127.5	206.3	67.5	138.8	7.40	143.5	88.5	133.8	52.4	81.4
3500	13.51	186.75	131.0	202.5	75.1	127.4	7.39	146.25	91.75	129.9	60.2	69.7
4000	13.80	190.0	135.0	197.9	84.1	113.8	6.77	149.5	95.25	125.3	69.4	55.9
4500	13.40	193.75	139.25	192.7	94.5	98.2	5.46	152.5	98.75	120.0	79.9	40.1
5000	12.20	197.75	143.75	186.8	106.3	80.5	3.37	156.0	103.0	114.1	91.9	22.2
5500	10.15	201.75	148.25	180.3	119.4	60.9	0.40	159.5	106.25	107.5	105.0	2.5
170 Degrees Arc of Contact.												
250	1.00	185.25	129.5	204.2	71.6	132.6	0.67	142.0	86.25	136.1	47.8	88.3
500	2.18	182.0	125.75	208.0	64.1	143.9	1.45	140.0	84.0	138.6	42.9	95.7
750	3.40	180.75	123.5	209.9	60.2	149.7	2.26	139.0	82.5	139.8	40.5	99.3
1000	4.61	180.0	123.0	210.7	58.7	152.0	3.04	139.0	82.25	140.1	39.8	100.3
1200	5.57	179.5	122.5	211.1	57.8	153.3	3.65	138.75	82.25	140.2	39.6	100.6
1400	6.49	179.5	122.5	211.0	58.0	153.0	4.23	139.0	82.50	139.9	40.2	99.7
1600	7.38	180.0	123.0	210.8	58.5	152.3	4.78	139.0	83.0	139.5	41.0	98.5
1800	8.24	180.25	123.25	210.4	59.3	151.1	5.29	139.50	83.5	139.0	42.0	97.0
2000	9.05	180.75	123.75	209.8	60.4	149.4	5.75	140.0	84.25	138.3	43.4	94.9
2500	10.83	182.5	126.0	207.7	64.7	143.0	6.66	142.0	86.5	136.0	48.1	87.9
3000	12.21	184.75	129.0	204.8	70.5	134.3	7.16	144.25	89.25	132.9	54.2	78.7
3500	13.08	187.75	132.25	201.1	77.8	123.3	7.15	146.75	92.25	129.1	61.8	67.4
4000	13.37	191.0	136.0	196.8	86.5	110.3	6.56	149.75	95.75	124.7	70.6	54.1
4500	12.97	194.5	140.25	191.7	96.6	95.1	5.29	152.75	99.0	119.6	80.8	38.8
5000	11.81	198.5	144.5	186.0	108.0	78.0	3.26	156.25	103.25	113.8	92.3	21.5
5500	9.83	202.25	148.75	179.7	120.7	59.0	0.39	159.5	106.25	107.4	105.1	2.3

Table I—*Continued*

Horsepower and Tensions of Belts

Velocity of Belt, Ft. per min.	Machine Belts.						Countershaft Belts.					
	Horsepower.	Initial Tension, lb. t_m.	Minimum Tension, lb. t_0.	Tension on Tight Side, lb. t_1.	Tension on Slack Side, lb. t_2.	Effective pull, lb., t_1-t_2.	Horsepower.	Initial tension, lb. t_m.	Minimum Tension, lb. t_0.	Tension of Tight Side, lb. t_1.	Tension on Slack Side, lb. t_2.	Effective Pull, lb., t_1-t_2.
160 Degrees Arc of Contact.												
250	0.96	186.75	131.0	202.3	75.4	126.9	0.64	143.0	87.5	134.8	50.3	84.5
500	2.09	183.75	127.75	206.1	67.8	138.3	1.39	141.0	85.25	137.3	45.5	91.8
750	3.27	182.25	125.75	207.9	64.1	143.8	2.17	140.0	84.0	138.6	42.8	95.8
1000	4.43	181.75	125.0	208.7	62.5	146.2	2.92	140.0	83.5	138.8	42.4	96.4
1200	5.36	181.25	124.5	209.1	61.8	147.3	3.51	139.75	83.5	138.9	42.2	96.7
1400	6.25	181.25	124.5	209.1	61.9	147.2	4.07	140.0	83.75	138.7	42.7	96.0
1600	7.11	181.25	125.0	208.9	62.3	146.6	4.60	140.25	84.25	138.3	43.4	94.9
1800	7.94	181.75	125.25	208.5	63.0	145.5	5.09	140.50	84.75	137.8	44.4	93.4
2000	8.72	182.25	126.75	208.0	64.1	143.9	5.54	141.0	85.25	137.1	45.7	91.4
2500	10.44	184.0	127.75	205.9	68.1	137.8	6.42	142.75	87.50	134.9	50.2	84.7
3000	11.77	186.0	130.5	203.2	73.7	129.5	6.90	144.75	90.0	132.0	56.1	75.9
3500	12.61	189.0	133.5	199.7	80.7	119.0	6.90	147.25	93.75	128.3	63.3	65.0
4000	12.89	192.0	137.0	195.5	89.1	106.4	6.33	150.25	96.25	124.1	71.9	52.2
4500	12.52	195.25	141.0	190.6	98.8	91.8	5.11	153.25	99.5	119.2	81.7	37.5
5000	11.40	199.0	145.0	185.1	109.8	75.2	3.15	156.5	103.5	113.6	92.8	20.8
5500	9.49	202.75	149.25	179.0	122.1	56.9	0.38	159.5	106.25	107.4	105.2	2.2
150 Degrees Arc of Contact.												
250	0.92	188.25	132.75	200.3	79.4	120.9	0.61	143.75	88.75	133.5	53.0	80.5
500	2.00	185.5	129.5	203.9	72.1	131.8	1.33	142.0	86.5	135.9	48.2	87.7
750	3.12	184.0	128.0	205.8	68.4	137.4	2.07	141.0	85.5	137.0	45.9	91.1
1000	4.24	183.25	127.0	206.7	66.7	140.0	2.80	141.0	85.0	137.4	45.1	92.3
1200	5.13	183.0	126.5	207.1	65.9	141.2	3.37	140.75	85.0	137.5	44.9	92.6
1400	5.99	183.0	126.5	207.1	65.9	141.2	3.90	141.0	85.0	137.3	45.3	92.0
1600	6.82	183.0	127.0	206.9	66.3	140.6	4.41	141.25	85.5	137.0	46.0	91.0
1800	7.62	183.5	127.0	206.5	66.9	139.6	4.89	141.5	86.0	136.5	46.9	89.6
2000	8.37	183.75	127.5	206.0	68.0	138.0	5.32	142.0	86.5	135.9	48.2	87.7
2500	10.02	185.5	129.5	204.1	71.8	132.3	6.16	143.5	88.5	133.8	52.5	81.3
3000	11.31	187.5	132.0	201.5	77.1	124.4	6.63	145.5	90.75	131.0	58.1	72.9
3500	12.12	190.0	135.0	198.1	83.8	114.3	6.63	148.0	93.5	127.5	65.0	62.5
4000	12.39	193.0	138.25	194.1	91.9	102.2	6.08	150.5	96.75	123.4	73.2	50.2
4500	12.03	196.0	142.0	189.4	101.2	88.2	4.91	153.5	99.75	118.7	82.7	36.0
5000	10.96	199.75	145.75	184.1	111.8	72.3	3.02	156.5	103.75	113.3	93.3	20.0
5500	9.12	203.25	149.75	178.2	123.5	54.7	0.36	159.75	106.25	107.4	105.2	2.2

Table I—*Continued*

Horsepower and Tensions of Belts

Velocity of Belt, Ft. per min.	Machine Belts. Horsepower.	Machine Belts. Initial Tension, lb. t_m.	Machine Belts. Minimum Tension, lb. t_0.	Machine Belts. Tension on Tight Side, lb. t_1.	Machine Belts. Tension on Slack Side, lb. t_2.	Machine Belts. Effective pull, lb. t_1-t_2.	Countershaft Belts. Horsepower.	Countershaft Belts. Initial Tension, lb. t_m.	Countershaft Belts. Minimum Tension, lb. t_0.	Countershaft Belts. Tension on Tight Side, lb. t_1.	Countershaft Belts. Tension on Slack Side, lb. t_2.	Countershaft Belts. Effective Pull, lb., t_1-t_2.

140 Degrees Arc of Contact.

250	0.87	190.0	134.75	198.2	83.6	114.6	0.58	144.75	90.0	132.1	55.8	76.3
500	1.90	187.0	131.5	201.8	76.5	125.3	1.26	143.25	87.75	134.4	51.1	83.3
750	2.98	185.75	130.0	203.7	72.7	131.0	1.97	142.25	86.75	135.6	48.8	86.8
1000	4.04	185.0	129.0	204.5	71.1	133.4	2.67	142.0	86.50	136.0	48.0	88.0
1200	4.90	184.75	128.5	204.9	70.3	134.6	3.21	141.75	86.50	136.1	47.8	88.3
1400	5.71	184.75	128.5	204.9	70.2	134.7	3.73	142.0	86.50	135.9	48.1	87.8
1600	6.51	185.0	129.0	204.7	70.5	134.2	4.21	142.25	86.75	135.6	48.7	86.9
1800	7.27	185.0	129.25	204.4	71.1	133.3	4.67	142.5	87.25	135.2	49.7	85.5
2000	7.99	185.5	129.5	204.0	72.1	131.9	5.08	143.0	87.75	134.6	50.8	83.8
2500	9.57	187.0	131.5	202.1	75.7	126.4	5.89	144.5	89.50	132.6	54.9	77.7
3000	10.81	189.0	133.5	199.6	80.8	118.8	6.34	146.25	91.75	129.9	60.2	69.7
3500	11.59	191.25	136.5	196.4	87.2	109.2	6.34	148.5	94.25	126.6	66.8	59.8
4000	11.85	194.0	139.25	193.2	93.5	97.7	5.82	151.0	97.50	122.7	74.7	48.0
4500	11.50	197.0	143.0	188.1	103.8	84.4	4.69	153.75	100.25	118.1	83.7	34.4
5000	10.48	200.25	146.5	183.1	113.9	69.2	2.89	156.75	104.0	113.0	93.9	19.1
5500	8.73	203.75	150.25	177.5	125.1	52.4	0.35	159.75	106.25	107.4	105.3	2.1

130 Degrees Arc of Contact.

250	0.82	191.5	136.75	196.0	87.9	108.1	0.55	145.75	91.25	130.7	58.7	72.0
500	1.79	189.0	133.5	199.5	81.0	118.5	1.20	144.25	89.25	132.9	54.2	78.8
750	2.82	187.5	132.0	201.5	77.1	124.4	1.87	143.25	88.25	134.1	51.8	82.3
1000	3.83	187.0	131.25	202.2	75.7	126.5	2.53	143.0	87.75	134.5	51.1	83.4
1200	4.64	186.5	130.75	202.6	74.9	127.7	3.05	143.0	87.75	134.6	50.8	83.8
1400	5.42	186.5	130.75	202.6	74.8	127.8	3.54	143.0	87.75	134.4	51.1	83.3
1600	6.18	186.75	131.0	202.5	75.1	127.4	4.00	143.5	88.25	134.2	51.7	82.5
1800	6.91	187.0	131.25	202.2	75.6	126.6	4.43	143.5	88.50	133.8	52.5	81.3
2000	7.59	187.25	131.75	201.8	76.5	125.3	4.82	144.0	89.0	133.2	53.6	79.6
2500	9.10	188.5	133.25	200.1	79.9	120.2	5.60	145.5	90.5	131.3	57.5	73.8
3000	10.27	190.5	135.25	197.7	84.7	113.0	6.02	147.0	92.75	128.8	62.5	66.3
3500	11.02	192.5	137.75	194.6	90.7	103.9	6.03	149.25	95.0	125.6	68.8	56.8
4000	11.26	195.0	140.5	191.0	98.0	93.0	5.53	151.5	98.25	121.9	76.2	45.6
4500	10.94	198.0	144.0	186.8	106.5	80.3	4.46	154.25	100.50	117.6	84.9	32.7
5000	9.97	201.0	147.25	182.0	116.1	65.8	2.75	156.75	104.00	112.7	94.6	18.2
5500	8.30	204.25	150.75	176.6	126.8	49.8	0.33	159.75	106.25	107.3	105.3	2.0

Table I—*Continued*

Horsepower and Tensions of Belts

Velocity of Belt, Ft. per min.	Machine Belts.						Countershaft Belts.					
	Horsepower.	Initial tension, lb. t_m.	Minimum Tension, lb. t_0.	Tension of Tight Side, lb. t_1.	Tension of Slack Side, lb. t_2.	Effective Pull, lb. t_1-t_2.	Horsepower.	Initial Tension, lb. t_m.	Minimum Tension, lb. t_0.	Tension on Tight Side, lb. t_1.	Tension of Slack Side, lb. t_2.	Effective Pull, lb. t_1-t_2.
120 Degrees Arc of Contact.												
250	0.77	193.0	138.5	193.8	92.5	101.3	0.51	146.75	92.5	129.2	61.7	67.5
500	1.68	190.75	135.75	197.1	85.8	111.3	1.12	145.5	90.5	131.2	57.5	73.7
750	2.65	189.5	134.25	198.9	82.2	116.7	1.76	144.5	89.5	132.5	55.1	77.4
1000	3.61	189.5	133.0	199.7	80.6	119.1	2.38	144.25	89.25	132.9	54.3	78.6
1200	4.37	188.5	132.75	200.1	79.9	120.2	2.87	144.0	89.25	133.0	54.1	78.9
1400	5.11	188.5	132.75	200.2	79.7	120.5	3.34	144.0	89.25	132.9	54.2	78.7
1600	5.83	188.5	133.25	200.1	79.9	120.2	3.77	144.5	89.5	132.6	54.8	77.8
1800	6.53	188.5	133.25	199.9	80.2	119.8	4.18	144.75	89.75	132.2	55.5	76.7
2000	7.17	189.0	133.5	199.4	81.2	118.2	4.55	145.0	90.25	131.7	56.6	75.1
2500	8.60	190.25	135.0	197.8	84.4	113.5	5.29	146.25	91.75	129.9	60.2	69.7
3000	9.71	191.75	137.0	195.6	88.8	106.8	5.69	148.0	93.5	127.5	64.9	62.6
3500	10.41	193.75	139.25	192.7	94.5	98.2	5.70	149.75	95.75	124.6	70.9	53.7
4000	10.65	196.25	142.0	189.3	101.4	87.9	5.23	152.0	99.0	121.0	77.9	43.1
4500	10.35	199.0	145.0	185.3	109.4	75.9	4.22	154.5	101.0	117.0	86.0	31.0
5000	9.43	201.75	148.0	180.7	118.5	62.2	2.60	157.0	104.25	112.4	95.2	17.2
5500	7.85	204.75	151.25	175.7	128.6	47.1	0.32	159.75	106.25	107.3	105.4	1.9

Stretch of Belts.

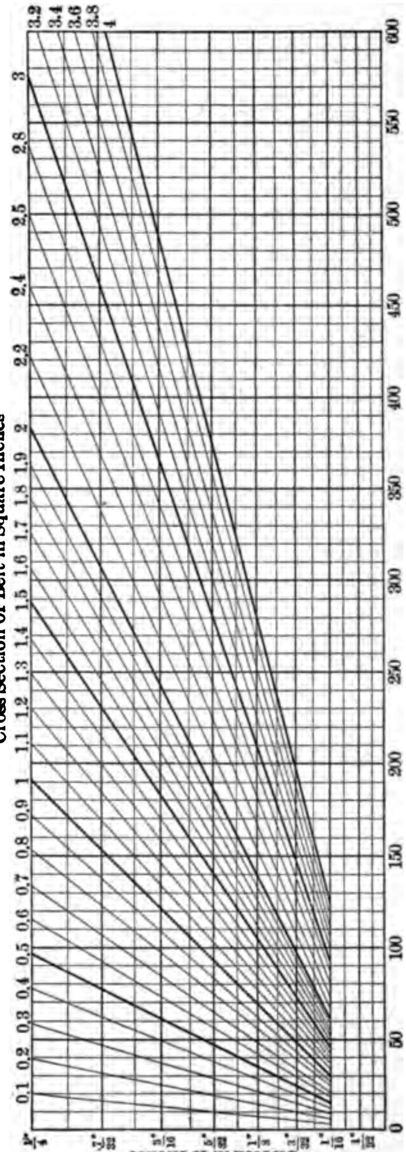


FIG. 32.—STRETCH OF BELT REQUIRED FOR A GIVEN TENSION.

Table II

Arc of Contact of Belt on Smaller Pulley, Degrees

Center to Center Distance of Pulleys.		Difference in Diameter of Pulleys, In.									
		1	2	3	4	5	6	7	8	9	10
ft	in.										
2	0	178	175	173	170	168	166	163	161	158	156
2	6	178	176	174	172	170	169	167	165	163	161
3	0	178	177	175	174	172	170	169	167	166	164
3	6	179	177	176	175	173	172	170	169	168	166
4	0	179	178	176	175	174	173	172	170	169	168
4	6	179	178	177	176	175	174	173	172	170	169
5	0	179	178	177	176	175	174	173	172	171	170
5	6	179	178	177	177	176	175	174	173	172	171
6	0	179	178	177	177	176	175	174	174	173	172
6	6	179	179	178	177	176	176	175	174	173	173
7	0	179	179	178	177	177	176	175	175	174	173
7	6	179	179	178	177	177	176	176	175	174	174
8	0	179	179	178	178	177	177	176	175	175	174
8	6	179	179	178	178	177	177	176	176	175	174
9	0	179	179	178	178	177	177	176	176	175	175
9	6	179	179	178	178	177	177	176	176	175	175
10	0	180	179	179	178	178	177	177	176	176	175
10	6	180	179	179	178	178	177	177	176	176	175
11	0	180	179	179	178	178	177	177	177	176	176
11	6	180	179	179	178	178	178	177	177	176	176
12	0	180	179	179	178	178	178	177	177	176	176
12	6	180	179	179	178	178	178	177	177	177	176
13	0	180	179	179	179	178	178	177	177	177	176
13	6	180	179	179	179	178	178	178	177	177	176
14	0	180	179	179	179	178	178	178	177	177	177
14	6	180	179	179	179	178	178	178	177	177	177
15	0	180	179	179	179	178	178	178	177	177	177
15	6	180	179	179	179	179	178	178	178	177	177
16	0	180	179	179	179	179	178	178	178	177	177
16	6	180	179	179	179	179	178	178	178	177	177
17	0	180	179	179	179	179	178	178	178	177	177
17	6	180	179	179	179	179	178	178	178	178	177
18	0	180	179	179	179	179	178	178	178	178	177
18	6	180	179	179	179	179	178	178	178	178	177
19	0	180	179	179	179	179	178	178	178	178	177
19	6	180	180	179	179	179	179	178	178	178	178
20	0	180	180	179	179	179	179	178	178	178	178

Table II—*Continued*

Arc of Contact of Belt on Smaller Pulley, Degrees

Center to Center Distance of Pulleys.		Difference in Diameter of Pulleys, In.									
		11	12	13	14	15	16	17	18	19	20
ft.	in.										
2	0	153	151	149	146	144	141	139	136	—	—
2	6	159	157	155	153	151	149	147	145	143	141
3	0	162	161	159	158	156	155	153	151	149	148
3	6	165	164	162	161	159	158	157	155	154	152
4	0	167	166	164	163	162	161	160	158	157	156
4	6	168	167	166	165	164	163	162	161	160	159
5	0	169	169	168	167	166	165	164	163	162	161
5	6	170	170	169	168	167	166	165	164	163	163
6	0	171	170	170	169	168	167	166	166	165	164
6	6	172	171	170	170	169	168	167	167	166	165
7	0	172	172	171	170	170	169	168	168	167	166
7	6	173	172	172	171	170	170	169	169	168	167
8	0	174	173	173	172	171	171	170	170	169	169
8	6	174	173	173	172	172	171	170	170	169	169
9	0	174	174	173	173	172	172	171	170	170	169
9	6	174	174	173	173	172	172	171	171	170	170
10	0	175	174	174	173	173	172	172	171	171	170
10	6	175	175	174	174	173	173	172	172	171	171
11	0	175	175	174	174	173	173	173	172	172	171
11	6	175	175	175	174	174	173	173	173	172	172
12	0	176	175	175	174	174	174	173	173	172	172
12	6	176	175	175	175	174	174	174	173	173	172
13	0	176	176	175	175	174	174	174	173	173	173
13	6	176	176	175	175	175	174	174	174	173	173
14	0	176	176	176	175	175	175	174	174	174	173
14	6	176	176	176	175	175	175	174	174	174	173
15	0	176	176	176	176	175	175	175	174	174	174
15	6	177	176	176	176	175	175	175	174	174	174
16	0	177	176	176	176	176	175	175	175	174	174
16	6	177	177	176	176	176	175	175	175	175	174
17	0	177	177	176	176	176	176	175	175	175	174
17	6	177	177	176	176	176	176	175	175	175	175
18	0	177	177	177	176	176	176	175	175	175	175
18	6	177	177	177	176	176	176	176	175	175	175
19	0	177	177	177	176	176	176	176	175	175	175
19	6	177	177	177	177	176	176	176	176	175	175
20	0	177	177	177	177	176	176	176	176	175	175

Table II—*Continued*

Arc of Contact of Belt on Smaller Pulley, Degrees

Center to Center Distance of Pulleys.		Difference in Diameter of Pulleys, In.									
		21	22	23	24	25	26	27	28	29	30
ft.	in.										
2	0	—	—	—	—	—	—	—	—	—	—
2	6	139	137	135	133	—	—	—	—	—	—
3	0	146	145	143	141	139	138	136	134	132	131
3	6	151	150	148	147	145	144	143	141	140	138
4	0	155	154	152	151	150	149	147	146	145	144
4	6	158	156	155	154	153	152	151	150	149	148
5	0	160	159	158	157	156	155	154	153	152	151
5	6	162	161	160	159	158	157	156	156	155	154
6	0	163	162	162	161	160	159	158	158	157	156
6	6	165	164	163	162	162	161	160	159	158	158
7	0	166	165	164	164	163	162	162	161	160	159
7	6	167	166	165	165	164	163	163	162	161	161
8	0	168	167	167	166	166	165	164	164	163	163
8	6	168	168	167	166	166	165	165	164	164	163
9	0	169	168	168	167	167	166	166	165	165	164
9	6	169	169	168	168	167	167	166	166	165	165
10	0	170	170	169	169	168	168	167	167	166	166
10	6	170	170	170	169	169	168	168	167	167	166
11	0	171	170	170	170	169	169	168	168	167	167
11	6	171	171	170	170	170	169	169	168	168	168
12	0	172	171	171	170	170	170	169	169	168	168
12	6	172	172	171	171	170	170	170	169	169	169
13	0	172	172	172	171	171	170	170	170	169	169
13	6	172	172	172	172	171	171	170	170	170	169
14	0	173	173	172	172	171	171	171	170	170	170
14	6	173	173	172	172	172	171	171	171	170	170
15	0	173	173	173	172	172	172	171	171	171	170
15	6	174	173	173	173	172	172	172	171	171	171
16	0	174	173	173	173	173	172	172	172	171	171
16	6	174	174	173	173	173	172	172	172	172	171
17	0	174	174	174	173	173	173	172	172	172	172
17	6	174	174	174	173	173	173	173	172	172	172
18	0	174	174	174	174	173	173	173	173	172	172
18	6	175	174	174	174	174	173	173	173	173	172
19	0	175	174	174	174	174	173	173	173	173	172
19	6	175	175	174	174	174	174	173	173	173	173
20	0	175	175	175	174	174	174	174	173	173	173

Table II—*Continued*

Arc of Contact of Belt on Smaller Pulley, Degrees

Center to Center Distance of Pulleys.		Difference in Diameter of Pulleys, In.									
		31	**32**	**33**	**34**	**35**	**36**	**37**	**38**	**39**	**40**
ft.	in.										
2	0	—	—	—	—	—	—	—	—	—	—
2	6	—	—	—	—	—	—	—	—	—	—
3	0	—	—	—	—	—	—	—	—	—	—
3	6	137	135	134	132	131	129	—	—	—	—
4	0	142	141	140	139	137	136	135	133	132	131
4	6	147	146	144	143	142	141	140	139	138	137
5	0	150	149	148	147	146	145	144	143	142	141
5	6	153	152	151	150	149	148	147	147	146	145
6	0	155	154	154	153	152	151	150	149	149	148
6	6	157	156	156	155	154	153	153	152	151	150
7	0	159	158	157	156	156	155	155	154	153	152
7	6	160	160	159	158	158	157	156	156	155	154
8	0	161	161	160	160	159	159	158	157	157	156
8	6	163	162	161	161	160	160	159	159	158	157
9	0	164	163	162	162	161	161	160	160	159	159
9	6	164	164	163	163	162	162	161	161	160	160
10	0	165	165	164	164	163	163	162	162	161	161
10	6	166	165	165	165	164	164	163	163	162	162
11	0	167	166	166	165	165	164	164	163	163	163
11	6	167	167	166	166	165	165	165	164	164	163
12	0	168	167	167	166	166	166	165	165	164	164
12	6	168	168	167	167	167	166	166	165	165	165
13	0	169	168	168	168	167	167	166	166	166	165
13	6	169	169	168	168	168	167	167	167	166	166
14	0	169	169	169	168	168	168	167	167	167	166
14	6	170	169	169	169	168	168	168	167	167	167
15	0	170	170	170	169	169	169	168	168	168	167
15	6	170	170	170	170	169	169	169	168	168	168
16	0	171	170	170	170	170	169	169	169	168	168
16	6	171	171	170	170	170	170	169	169	169	168
17	0	171	171	171	170	170	170	170	169	169	169
17	6	172	171	171	171	170	170	170	170	169	169
18	0	172	172	171	171	171	170	170	170	170	169
18	6	172	172	171	171	171	171	170	170	170	170
19	0	172	172	172	171	171	171	171	170	170	170
19	6	172	172	172	172	171	171	171	171	170	170
20	0	173	172	172	172	172	171	171	171	171	170

Table II—*Continued*

Arc of Contact of Belt on Smaller Pulley, Degrees

Center to Center Distance of Pulleys.		Difference in Diameter of Pulleys, In.									
		41	42	43	44	45	46	47	48	49	50
ft.	in.										
2	0	—	—	—	—	—	—	—	—	—	—
2	6	—	—	—	—	—	—	—	—	—	—
3	0	—	—	—	—	—	—	—	—	—	—
3	6	—	—	—	—	—	—	—	—	—	—
4	0	129	128	127	126	—	—	—	—	—	—
4	6	135	134	133	132	131	130	128	127	—	—
5	0	140	139	138	137	136	135	134	133	132	131
5	6	144	143	142	141	140	139	138	137	136	136
6	0	147	146	145	144	144	143	142	141	140	139
6	6	150	149	148	147	146	146	144	144	143	143
7	0	152	151	150	149	149	148	148	147	146	145
7	6	154	153	152	152	151	150	150	149	148	148
8	0	156	155	154	154	153	152	152	151	151	150
8	6	157	156	156	155	155	154	153	153	152	152
9	0	158	158	157	156	156	155	155	154	154	153
9	6	159	159	158	158	157	157	156	156	155	155
10	0	160	160	159	159	158	158	157	157	156	156
10	6	161	161	160	160	159	159	159	158	158	157
11	0	162	162	161	161	160	160	159	159	159	158
11	6	163	163	162	162	161	161	160	160	160	159
12	0	164	163	163	162	162	162	161	161	160	160
12	6	164	164	164	163	163	162	162	162	161	161
13	0	165	165	164	164	163	163	163	162	162	162
13	6	165	165	165	164	164	164	163	163	163	162
14	0	166	166	165	165	165	164	164	164	163	163
14	6	166	166	166	166	165	165	164	164	164	164
15	0	167	167	166	166	166	165	165	165	164	164
15	6	167	167	167	166	166	166	166	165	165	165
16	0	168	167	167	167	167	166	166	166	165	165
16	6	168	168	168	167	167	167	166	166	166	166
17	0	168	168	168	168	167	167	167	166	166	166
17	6	169	169	168	168	168	167	167	167	167	166
18	0	169	169	169	168	168	168	168	167	167	167
18	6	169	169	169	169	168	168	168	168	167	167
19	0	170	169	169	169	169	168	168	168	168	167
19	6	170	170	169	169	169	169	169	168	168	168
20	0	170	170	170	170	169	169	169	169	168	168

Table II—*Continued*

Arc of Contact of Belt on Smaller Pulley, Degrees

Center to Center Distance of Pulleys.		Difference in Diameter of Pulleys, In.									
		51	52	53	54	55	56	57	58	59	60
ft.	in.										
2	0	—	—	—	—	—	—	—	—	—	—
2	6	—	—	—	—	—	—	—	—	—	—
3	0	—	—	—	—	—	—	—	—	—	—
3	6	—	—	—	—	—	—	—	—	—	—
4	0	—	—	—	—	—	—	—	—	—	—
4	6	—	—	—	—	—	—	—	—	—	—
5	0	130	129	128	127	125	124	123	122	121	120
5	6	135	134	133	132	131	130	129	128	127	126
6	0	139	138	137	136	135	134	133	133	132	131
6	6	142	141	140	139	139	138	137	136	136	135
7	0	145	144	143	143	142	141	140	140	139	138
7	6	147	146	146	145	144	144	143	142	142	141
8	0	149	149	148	148	147	146	146	145	145	144
8	6	151	150	150	149	149	148	148	147	146	146
9	0	153	152	152	151	150	150	149	149	148	148
9	6	154	154	153	153	152	152	151	151	150	150
10	0	156	155	155	154	154	153	153	152	152	151
10	6	157	156	156	155	155	154	154	153	153	153
11	0	158	157	157	156	156	155	155	155	154	154
11	6	159	158	158	157	157	157	156	156	155	155
12	0	160	159	159	158	158	158	157	157	156	156
12	6	160	160	160	159	159	158	158	158	157	157
13	0	161	161	160	160	160	159	159	159	158	158
13	6	162	162	161	161	160	160	160	159	159	159
14	0	163	162	162	162	161	161	160	160	160	159
14	6	163	163	162	162	162	161	161	161	160	160
15	0	164	163	163	163	162	162	162	162	161	161
15	6	164	164	164	163	163	163	162	162	162	161
16	0	165	164	164	164	164	163	163	163	162	162
16	6	165	165	165	164	164	164	163	163	163	163
17	0	166	165	165	165	165	164	164	164	163	163
17	6	166	166	166	165	165	165	164	164	164	164
18	0	166	166	166	166	165	165	165	165	164	164
18	6	167	167	166	166	166	166	165	165	165	164
19	0	167	167	167	166	166	166	166	165	165	165
19	6	168	167	167	167	167	166	166	166	166	165
20	0	168	168	167	167	167	167	166	166	166	166

Table III

Velocity of Belt, Feet per Minute, for Different Pulley Diameters and Revolutions per Minute

R.P.M.	Diameter of Pulley, In.											
	3	3 1/2	4	4 1/2	5	5 1/2	6	6 1/2	7	7 1/2	8	8 1/2
120	—	—	—	—	—	—	—	—	—	—	251.3	267.0
130	—	—	—	—	—	—	—	—	—	255.2	272.2	289.3
140	—	—	—	—	—	—	—	—	256.5	274.8	293.2	311.5
150	—	—	—	—	—	—	—	255.2	274.8	294.5	314.1	333.8
160	—	—	—	—	—	—	251.4	272.2	293.1	314.1	335.0	356.0
170	—	—	—	—	—	—	267.1	289.2	311.4	333.7	356.0	378.3
180	—	—	—	—	—	259.2	282.8	306.2	329.8	353.3	376.9	400.5
190	—	—	—	—	—	273.6	298.5	323.2	348.1	373.0	397.9	422.8
200	—	—	—	—	261.8	288.0	314.2	340.2	366.4	392.6	418.8	445.0
210	—	—	—	—	274.9	302.4	329.9	357.2	384.7	412.2	439.7	467.3
220	—	—	—	259.2	288.0	316.8	345.6	374.2	403.0	431.9	460.7	489.5
230	—	—	—	271.0	301.1	331.2	361.3	391.2	421.4	451.5	481.6	511.8
240	—	—	250.9	282.7	314.2	345.6	377.0	408.2	439.7	471.1	502.6	534.0
250	—	—	261.4	294.5	327.3	360.0	392.8	425.3	458.0	490.8	523.5	556.3
260	—	—	271.8	306.3	340.3	374.4	408.5	442.3	476.3	510.4	544.4	578.5
270	—	—	282.3	318.1	353.4	388.8	424.2	459.3	494.6	530.0	565.4	600.8
280	—	256.6	292.7	329.8	366.5	403.2	439.9	476.3	513.0	549.6	586.3	623.0
290	—	265.8	303.2	341.6	379.6	417.6	455.6	493.3	531.3	569.3	607.3	645.3
300	—	275.0	313.7	353.4	392.7	432.0	471.3	510.3	549.6	588.9	628.2	667.5
320	251.3	293.3	334.6	377.0	418.9	460.8	502.7	544.3	586.2	628.2	670.1	712.0
340	267.0	311.6	355.5	400.5	445.1	489.6	534.1	578.3	622.9	667.4	712.0	756.5
360	282.7	330.0	376.4	424.1	471.2	518.4	565.6	612.4	659.5	706.7	753.8	801.0
380	298.4	348.3	397.3	447.6	497.4	547.2	597.0	646.4	696.2	745.9	795.7	845.5
400	314.2	366.6	418.2	471.2	523.6	576.0	628.4	680.4	732.8	785.2	837.6	890.0
425	333.8	389.6	444.4	500.6	556.3	612.0	667.7	722.9	778.6	834.3	889.9	945.6
450	353.4	412.5	470.5	530.0	589.0	648.0	707.0	765.4	824.4	883.4	942.3	1001.2
475	373.1	435.4	496.6	559.4	621.8	684.0	746.2	807.9	870.2	932.4	994.6	1056.8
500	392.7	458.3	522.8	589.0	654.5	720.0	785.5	850.5	916.0	981.5	1047.0	1112.5
525	412.3	481.2	549.0	618.4	687.2	756.0	824.8	893.0	961.8	1030.6	1099.3	1168.1
550	432.0	504.1	575.1	647.9	719.9	792.0	864.0	935.5	1007.6	1079.6	1151.7	1213.8
575	451.6	527.0	601.2	677.4	752.7	828.0	903.3	978.1	1053.4	1128.7	1204.0	1269.4
600	471.2	550.0	627.3	706.8	785.4	864.0	942.6	1020.6	1099.2	1177.8	1256.4	1335.0
650	510.5	595.8	679.5	765.7	850.8	936.0	1021.2	1105.6	1190.8	1276.0	1370.1	1446.2
700	549.8	641.6	731.8	824.6	916.3	1008.0	1099.7	1190.7	1282.4	1374.1	1465.8	1557.5
750	589.1	687.4	784.1	883.4	981.7	1080.0	1178.3	1275.7	1374.0	1472.2	1570.5	1668.7
800	628.3	733.3	836.4	942.4	1047.2	1152.0	1256.8	1360.8	1465.6	1570.4	1675.2	1780.0
850	667.6	779.1	888.7	1001.3	1112.6	1224.0	1335.3	1445.0	1557.2	1668.5	1779.9	1891.2
900	706.9	824.9	940.9	1060.2	1178.1	1296.0	1413.9	1530.9	1648.8	1766.7	1884.6	2002.5
950	746.1	870.7	993.4	1119.1	1243.5	1368.0	1492.4	1615.9	1740.4	1864.8	1989.3	2113.7
1000	785.4	916.6	1045.5	1178.0	1309.0	1440.0	1571.0	1701.0	1832.0	1963.0	2094.0	2225.0
1100	863.9	1008.3	1050.0	1295.8	1439.9	1584.0	1728.1	1871.1	2015.2	2159.3	2303.4	2447.5
1200	942.5	1099.9	1254.6	1413.6	1570.8	1728.0	1885.2	2041.2	2198.4	2355.6	2512.8	2670.0
1300	1021.0	1191.6	1359.1	1531.4	1701.7	1872.0	2042.3	2211.3	2381.6	2551.9	2722.2	2892.5
1400	1099.6	1283.2	1463.7	1649.2	1832.6	2016.0	2199.4	2381.4	2564.8	2748.2	2931.6	3115.0
1500	1178.1	1374.9	1568.2	1767.0	1963.5	2160.0	2356.5	2551.5	2748.0	2944.5	3141.6	3337.5
1600	1256.6	1466.5	1672.8	1884.8	2094.4	2304.0	2513.6	2721.6	2931.2	3140.8	3350.4	3560.5
1700	1335.2	1558.2	1777.3	2002.6	2225.3	2448.0	2670.7	2891.7	3114.4	3337.1	3559.8	3782.0
1800	1413.7	1649.9	1881.9	2120.4	2356.4	2592.0	2827.8	3061.8	3297.6	3533.4	3769.2	4005.0

Table III—*Continued*

Velocity of Belt, Feet per Minute, for Different Pulley Diameters and Revolutions per Minute

R.P.M.	Diameter of Pulley, In.											
	9	9 1/2	10	10 1/2	11	11 1/2	12	12 1/2	13	13 1/2	14	14 1/2
70	—	—	—	—	—	—	—	—	—	—	256.6	265.7
80	—	—	—	—	—	—	251.3	261.8	272.5	282.7	293.2	303.7
90	—	—	—	—	259.2	271.0	282.7	294.5	306.5	318.1	329.9	341.6
100	—	—	261.8	274.9	288.0	301.1	314.2	327.2	340.6	353.4	366.5	379.6
110	259.2	273.6	288.0	302.4	316.8	331.2	345.5	359.9	374.7	388.7	403.2	417.6
120	282.7	298.4	314.2	329.9	345.6	361.8	376.9	392.6	408.7	424.1	439.8	455.5
130	306.3	323.3	340.3	357.4	374.4	391.4	408.3	425.4	442.8	459.4	476.5	493.5
140	329.8	348.2	366.5	384.9	403.2	421.5	439.7	458.1	476.8	494.8	513.1	531.4
150	353.4	373.1	392.7	412.4	432.0	451.7	471.2	490.8	510.9	530.1	549.8	569.4
160	377.0	397.9	418.9	439.8	460.8	481.8	502.6	523.5	545.0	565.4	586.4	607.4
170	400.5	422.8	445.1	467.3	489.6	511.9	534.0	556.2	579.0	600.8	623.1	645.3
180	424.1	447.7	471.2	494.8	518.4	542.0	565.4	589.0	613.1	636.1	659.7	683.3
190	447.6	472.5	497.4	522.3	547.2	572.1	597.8	621.7	647.1	671.5	696.4	721.2
200	471.2	497.4	523.6	549.8	576.0	602.2	628.2	654.4	681.2	706.8	733.0	759.2
210	494.8	522.3	549.8	577.3	604.8	632.3	659.6	687.1	715.3	742.1	769.7	797.2
220	518.3	547.1	576.0	604.8	633.6	662.4	691.0	719.8	749.3	777.5	806.3	835.1
230	541.9	572.0	602.1	632.3	662.4	692.5	722.4	752.6	783.4	812.8	843.0	873.1
240	565.4	596.9	628.3	659.8	691.2	722.6	753.8	785.3	817.4	848.2	879.6	911.0
250	589.0	621.8	654.5	687.3	720.0	752.8	785.3	818.0	851.5	883.5	916.3	949.0
260	612.6	646.6	680.7	714.7	748.8	782.9	816.7	850.7	855.6	918.8	952.9	987.0
270	636.1	671.5	706.9	742.2	777.6	813.0	848.1	883.4	919.6	954.2	989.6	1024.9
280	659.7	696.4	733.0	769.7	806.4	843.1	879.5	916.2	953.7	989.5	1026.2	1062.9
290	683.2	721.2	759.2	797.2	835.2	873.2	910.9	948.9	987.7	1024.9	1062.9	1100.8
300	706.8	746.1	785.4	824.7	864.0	903.3	942.3	981.6	1021.8	1060.2	1099.5	1138.8
320	753.9	795.8	837.8	879.7	921.6	963.5	1005.1	1047.0	1089.9	1130.9	1172.8	1214.7
340	801.0	845.6	890.1	934.7	979.2	1023.7	1067.9	1112.5	1158.0	1201.6	1246.1	1290.6
360	848.2	895.3	942.5	989.6	1036.8	1084.0	1130.8	1177.9	1226.2	1272.2	1319.4	1366.6
380	895.3	945.1	994.8	1044.6	1094.4	1144.2	1193.6	1243.4	1294.3	1342.9	1392.7	1442.5
400	942.4	994.8	1047.2	1099.6	1152.0	1204.4	1256.4	1308.8	1362.4	1413.6	1466.0	1518.0
425	1001.3	1057.0	1112.7	1168.3	1224.0	1279.7	1334.9	1390.6	1447.5	1501.9	1557.6	1613.3
450	1060.2	1119.1	1178.1	1237.1	1296.0	1354.9	1413.4	1472.4	1532.7	1590.3	1649.2	1708.2
475	1119.1	1181.3	1243.5	1305.8	1368.0	1430.2	1491.9	1555.2	1617.8	1678.6	1740.9	1803.1
500	1178.0	1243.5	1309.0	1374.5	1440.0	1505.5	1570.5	1636.0	1703.0	1767.0	1832.5	1898.0
525	1236.9	1305.7	1374.4	1443.2	1512.0	1580.8	1649.0	1717.8	1788.1	1855.3	1924.1	1992.9
550	1295.8	1367.9	1439.9	1511.9	1584.0	1656.1	1727.5	1799.6	1873.3	1943.7	2015.7	2087.8
575	1353.7	1430.1	1505.3	1580.7	1656.0	1731.2	1806.1	1881.4	1958.4	2032.0	2107.4	2182.7
600	1413.6	1492.2	1570.8	1649.4	1728.0	1806.6	1884.6	1963.2	2043.6	2120.4	2199.0	2277.6
650	1531.4	1616.6	1701.7	1786.8	1872.0	1957.2	2043.6	2126.8	2213.9	2297.1	2382.2	2467.4
700	1649.2	1740.9	1832.6	1924.3	2016.0	2107.7	2198.7	2290.4	2384.2	2473.8	2565.5	2657.2
750	1767.0	1865.3	1963.5	2061.7	2160.0	2258.3	2355.7	2454.0	2554.5	2650.5	2748.7	2847.0
800	1884.8	1989.6	2094.4	2199.2	2304.0	2408.8	2512.8	2617.6	2724.8	2827.2	2932.0	3036.8
850	2002.6	2113.9	2225.3	2336.6	2448.0	2559.3	2669.8	2781.2	2895.1	3003.9	3115.2	3226.6
900	2120.4	2238.3	2356.2	2474.1	2592.0	2709.9	2826.9	2944.8	3065.4	3180.6	3298.5	3416.4
950	2238.2	2362.6	2487.1	2611.5	2736.0	2860.5	2983.9	3108.4	3235.7	3357.3	3481.7	3606.2
1000	2356.0	2487.0	2618.0	2749.0	2880.0	3011.0	3141.6	3272.0	3406.0	3534.0	3665.0	3796.0
1100	2591.6	2735.7	2879.8	3023.9	3168.0	3312.1	3455.1	3599.2	3746.6	3887.4	4031.5	4175.6
1200	2827.2	2984.4	3141.6	3298.8	3456.0	3618.2	3769.2	3926.4	4087.2	4240.8	4398.0	4555.2
1300	3062.8	3233.1	3403.4	3573.7	3744.0	3914.3	4083.3	4253.6	4427.8	4594.2	4764.5	4934.8
1400	3298.4	3481.8	3665.2	3848.6	4032.0	4215.4	4397.4	4580.8	4768.4	4947.6	5131.0	5134.4
1500	3534.0	3730.5	3927.0	4123.5	4320.0	4516.5	4711.5	4908.0	5109.0	5301.0	5497.5	5694.0
1600	3769.6	3979.2	4188.8	4398.4	4608.0	4817.6	5025.6	5235.2	5449.6	5654.4	5864.0	—
1700	4005.2	4227.9	4450.6	4673.3	4896.0	5118.7	5339.7	5562.4	5790.2	—	—	—
1800	4240.8	4476.6	4712.4	4948.2	5184.0	5419.8	5653.8	5889.6	—	—	—	—

Table III—*Continued*

Velocity of Belt, Feet per Minute, for Different Pulley Diameters and Revolutions per Minute

[illegible]	Diameter of Pulley, In.											
	15	15½	16	16½	17	17½	18	18½	19	19½	20	21
50				—		—	—	—	—	255.3	261.8	274.9
60			[illegible]	[illegible]	[illegible]	274.9	282.7	290.6	298.4	306.3	314.2	329.9
70	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	320.7	329.8	339.0	348.2	357.4	366.5	384.9
80	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	377.0	387.4	397.9	408.4	418.9	439.8
90	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	424.1	435.9	447.7	459.5	471.2	494.8
100	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	471.2	484.3	497.4	510.5	523.6	549.8
110	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	518.3	532.7	547.1	561.6	576.0	604.8
120	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	565.4	581.2	596.9	612.6	628.3	659.8
130	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	612.6	629.6	646.6	663.7	680.7	714.7
140	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	659.7	678.0	696.4	714.7	733.0	769.7
150	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	706.8	726.5	746.1	765.8	785.4	824.7
160	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	753.9	774.9	795.8	816.8	837.8	879.7
170	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	801.0	823.3	845.6	867.9	890.1	934.7
180	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	848.2	871.7	895.3	918.9	942.5	989.6
190	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	895.3	920.2	945.1	970.0	994.8	1044.6
200	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	942.4	968.6	994.8	1021.0	1047.2	1099.6
210	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1017.0	1044.5	1072.1	1099.6	1154.6
220	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1065.5	1094.3	1123.1	1151.9	1209.6
230	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1083.8	1113.9	1144.0	1174.2	1204.3	1264.5
240	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1130.9	1162.3	1193.8	1225.2	1256.6	1319.5
250	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1210.8	1243.5	1276.3	1309.0	1374.5
260	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1259.2	1293.2	1327.3	1361.4	1429.5
270	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1307.6	1343.0	1378.4	1413.7	1484.5
280	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1356.0	1392.7	1429.4	1466.1	1539.4
290	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1404.5	1442.5	1480.5	1518.4	1594.4
300	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1452.9	1492.2	1531.5	1570.8	1649.4
320	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1549.8	1591.7	1633.6	1675.5	1759.4
340	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1645.6	1691.2	1735.7	1780.2	1869.3
360	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1743.5	1790.6	1837.8	1885.0	1979.3
380	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1840.3	1890.1	1939.9	1989.7	2089.2
400	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	1937.2	1989.6	2042.0	2094.4	2199.2
425	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2058.3	2113.9	2169.6	2225.3	2336.6
450	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2179.3	2238.3	2297.2	2356.2	2474.1
475	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2300.4	2362.6	2424.8	2487.1	2611.5
500	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2421.5	2487.0	2552.5	2618.0	2749.0
525	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2542.6	2611.3	2680.1	2748.9	2886.4
550	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2663.6	2735.7	2807.7	2879.8	3023.9
575	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	2784.7	2850.0	2935.3	3010.7	3161.3
600	2356.2	[illegible]	[illegible]	2591.4	[illegible]	2748.6	[illegible]	[illegible]	2984.4	3063.0	3141.6	3298.8
650	2552.6	2637.7	2722.2	2807.4	[illegible]	2977.6	[illegible]	3147.9	3233.1	3318.2	3403.4	3573.7
700	2748.9	2840.6	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3390.1	3481.8	3573.5	3665.2	3848.6
750	2945.3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	3730.5	3828.7	3827.0	4123.5
800	3141.6	3246.4	3350.4	3455.2	[illegible]	[illegible]	[illegible]	3874.4	3979.2	4084.0	4188.8	4398.4
850	3337.9	3449.3	3559.8	3671.1	[illegible]	[illegible]	4005.2	[illegible]	4227.9	4339.2	4450.6	4673.3
900	3534.3	3652.2	3769.2	3887.1	4005.9	4122.9	[illegible]	4356.7	4476.6	4594.5	4712.4	4948.2
950	3730.6	3855.1	3978.6	4103.0	4228.4	4351.9	4476.4	4600.8	4725.3	4849.7	4974.2	5223.1
1000	3927.0	4058.0	4188.0	4319.0	4451.0	4581.0	4712.0	4843.0	4974.0	5105.0	5236.0	5496.3
1100	4319.7	4463.8	4606.8	4750.9	4896.1	5039.1	5183.2	5327.3	5471.4	5615.5	5759.6	—
1200	4712.4	4869.6	5025.6	5182.8	5341.2	5497.2	5654.4	[illegible]	—	—	—	—
1300	5105.1	5275.4	5444.4	5614.7	5756.3	—	—	—	—	—	—	—
1400	5497.8	5681.2	5863.2	—	—	—	—	—	—	—	—	—
1500	5890.5	—	—	—	—	—	—	—	—	—	—	—
1600	—	—	—	—	—	—	—	—	—	—	—	—
1700	—	—	—	—	—	—	—	—	—	—	—	—
1800		—	—	—	—	—	—	—	—	—	—	—

Table III—*Continued*

Velocity of Belt, Feet per Minute, for Different Pulley Diameters and Revolutions per Minute

	Diameter of Pulley, In.											
	22	23	24	25	26	27	28	29	30	31	32	33
0	—	—	—	—	—	—	—	—	—		251.3	259.2
0	—	—	251.3	261.8	272.3	282.7	293.2	303.7	314.2	324.6	335.1	345.6
0	288.0	301.1	314.2	327.3	340.4	353.4	366.5	379.6	392.7	405.8	418.9	432.0
0	345.5	361.3	377.0	392.7	408.4	424.1	439.8	455.5	471.2	487.0	502.6	518.3
70	403.1	421.5	439.8	458.2	476.5	494.8	513.1	531.4	549.8	568.1	586.4	604.7
0	460.7	481.7	502.6	523.6	544.6	565.4	586.4	607.4	628.3	649.3	670.2	691.1
0	518.3	541.9	565.5	589.1	612.6	636.1	659.7	683.3	706.9	730.4	753.9	777.5
0	575.9	602.1	628.3	654.5	680.7	706.8	733.0	759.2	785.4	811.6	837.7	863.9
10	633.5	662.3	691.1	720.0	748.8	777.5	806.3	835.1	863.9	892.8	921.5	950.3
20	691.1	722.5	754.0	785.4	816.8	848.2	879.6	911.0	942.5	973.9	1005.2	1036.7
0	748.7	782.7	816.8	850.9	884.9	918.8	952.9	987.0	1021.0	1055.1	1089.0	1123.1
0	806.3	842.9	879.6	916.3	953.0	989.5	1026.2	1062.9	1099.6	1136.2	1172.8	1209.5
0	863.9	903.2	942.5	981.8	1021.1	1060.2	1099.5	1138.8	1178.1	1217.4	1256.6	1295.9
0	921.4	963.4	1005.3	1047.2	1089.1	1130.9	1172.8	1214.7	1256.6	1298.6	1340.3	1382.2
70	979.0	1023.6	1068.1	1112.7	1157.2	1201.6	1246.1	1290.6	1335.2	1379.7	1424.1	1468.6
0	1036.7	1083.8	1130.9	1178.1	1225.3	1272.2	1319.4	1365.6	1413.7	1460.9	1507.9	1555.0
0	1094.2	1144.0	1193.8	1243.6	1293.3	1342.9	1392.7	1442.5	1492.3	1542.0	1591.6	1641.4
0	1151.8	1204.2	1256.6	1309.0	1361.4	1413.6	1466.0	1518.4	1570.8	1623.2	1675.4	1727.8
10	1209.4	1264.4	1319.4	1374.5	1429.5	1484.3	1539.3	1594.3	1649.3	1704.4	1759.2	1814.2
20	1267.0	1324.6	1380.3	1439.9	1497.5	1555.0	1612.6	1670.2	1727.9	1785.5	1842.9	1900.6
0	1324.6	1384.8	1445.1	1505.4	1565.6	1625.6	1685.9	1746.2	1806.4	1866.7	1926.7	1987.0
0	1382.2	1445.0	1507.9	1570.8	1633.7	1696.3	1759.2	1822.1	1885.0	1947.8	2010.5	2073.4
0	1439.8	1505.3	1570.8	1635.3	1701.8	1767.0	1832.5	1898.0	1963.5	2029.0	2094.3	2159.8
0	1497.3	1565.5	1633.6	1701.7	1769.8	1837.7	1905.8	1973.9	2042.0	2110.2	2178.0	2246.1
70	1554.9	1625.7	1696.4	1767.2	1837.9	1908.4	1979.1	2049.8	2120.6	2191.3	2261.8	2332.5
0	1621.5	1685.9	1759.2	1832.6	1906.0	1979.0	2052.4	2125.8	2199.1	2272.5	2345.6	2418.9
0	1670.1	1746.1	1822.1	1898.1	1974.0	2049.7	2125.7	2201.7	2277.7	2353.6	2429.3	2505.3
0	1727.7	1806.3	1884.9	1963.5	2042.1	2120.4	2199.0	2277.6	2356.2	2434.8	2513.1	2591.7
20	1842.9	1926.7	2010.6	2094.4	2178.2	2261.8	2345.6	2429.4	2513.3	2597.1	2680.6	2764.5
0	1958.1	2047.1	2136.2	2225.3	2314.4	2403.1	2492.2	2581.3	2670.4	2759.4	2848.2	2937.3
0	2073.2	2167.6	2261.9	2356.2	2450.5	2544.5	2638.8	2733.1	2827.4	2921.8	3015.7	3110.0
0	2188.4	2288.0	2387.5	2487.1	2586.7	2685.8	2785.4	2885.0	2984.5	3084.1	3183.3	3282.8
0	2303.6	2408.4	2513.2	2618.0	2722.8	2827.2	2932.0	3036.8	3141.6	3246.4	3350.8	3455.6
5	2447.6	2558.9	2670.3	2781.6	2893.0	3003.9	3115.2	3226.6	3337.9	3449.3	3560.2	3671.5
0	2591.6	2709.4	2827.4	2945.2	3063.2	3180.6	3298.5	3416.4	3534.3	3652.2	3769.6	3887.4
5	2735.6	2859.9	2984.5	3108.9	3233.4	3357.3	3481.7	3606.2	3730.6	3855.1	3979.1	4103.4
0	2879.5	3010.5	3141.6	3272.5	3403.5	3534.0	3665.0	3796.0	3927.0	4058.0	4188.5	4319.5
5	3023.5	3161.0	3298.7	3436.1	3573.7	3710.7	3848.2	3985.8	4123.3	4260.9	4397.9	4535.4
0	3167.5	3311.5	3455.8	3599.8	3743.9	3887.4	4031.5	4175.6	4319.7	4463.8	4607.3	4751.3
5	3311.5	3462.0	3612.9	3763.4	3914.1	4064.1	4214.7	4365.4	4516.0	4666.7	4816.8	4967.3
0	3455.4	3612.6	3769.8	3927.0	4084.2	4240.8	4398.0	4555.2	4712.4	4869.6	5026.2	5183.2
0	3743.4	3913.6	4084.0	4254.2	4424.6	4594.2	4764.5	4934.8	5105.1	5275.4	5445.0	5615.0
0	4031.3	4214.7	4398.2	4581.5	4764.9	4947.6	5131.0	5314.4	5497.8	5681.2	5863.9	
0	4319.2	4515.7	4712.4	4908.7	5205.3	5301.0	5497.5	5694.0	5890.5	—	—	
0	4607.2	4816.8	5026.6	5236.0	5445.6	5654.4	5864.0	—	—	—	—	
0	4895.1	5117.8	5340.7	5563.2	5615.8	5831.1	—	—	—	—	—	
0	5183.1	5418.9	5654.8	5890.5	—	—	—	—	—	—	—	—
0	5471.0	5569.4	5969.0	—	—	—	—	—		—	—	—
0	5759.0	—	—	—	—	—	—	—	—	—	—	—
0	—	—	—	—	—	—	—	—	—	—	—	—

Table III—*Continued*

Velocity of Belt, Feet per Minute, for Different Pulley Diameters and Revolutions per Minute

R.P.M.	Diameter of Pulley, In.											
	34	35	36	38	40	42	44	46	48	50	52	54
20	—	—	—	—	—	—	—	—	251.3	261.8	272.3	282.7
30	267.0	274.9	282 8	298 4	314.2	329.9	345.6	361.3	377.0	392.7	408.4	424.1
40	356.0	366.5	377.0	397.9	418.9	439.8	460.8	481.7	502.6	523.6	544.5	565.5
50	445.1	458 2	471.3	497 4	523.6	549.8	576.0	602.2	628.3	654.5	680.7	706.9
60	534.1	549.8	565.5	596.9	628.3	659.7	691.1	722.6	754.0	785.4	816.8	848.2
70	623.1	641.4	659.8	696.4	733.0	769.7	806.3	843.0	879.6	916.3	952.9	989.6
80	712.1	733.0	754 0	795.8	837.8	879.6	921.5	963.4	1005.3	1047.2	1089.0	1131.0
90	801.1	824.7	848.3	895.3	942.5	989.6	1036.7	1083.9	1130.9	1178.1	1225.2	1272.3
100	890.1	916.3	942.5	994.8	1047.2	1099.5	1151.9	1204.3	1256.6	1309.0	1361.3	1413.7
110	979.1	1007.9	1036.8	1094.3	1151.9	1209.5	1267.1	1324.7	1382.3	1439.9	1497.4	1555.1
120	1068.1	1099.6	1131.0	1193.8	1256.6	1319.4	1382.3	1445.2	1507.9	1570.8	1633.6	1696.4
130	1157.1	1191.2	1225.3	1293.2	1361.4	1429.4	1497.5	1565.6	1633.6	1701.7	1769.7	1837.8
140	1264.1	1282.8	1319.5	1392.7	1466.1	1539.3	1612.7	1686.0	1769.2	1832.6	1905.8	1979.2
150	1335.2	1374.5	1413.8	1492.2	1570.8	1649.3	1727.9	1806.5	1894.9	1963.5	2042.0	2120.6
160	1424.2	1466.1	1508.0	1591.7	1675.5	1759.2	1843.0	1926.9	2020.6	2094.4	2178.1	2261.9
170	1513.2	1557.7	1602.3	1691.2	1780.2	1869.2	1958.2	2047.3	2146.2	2225.3	2314.2	2403.3
180	1602.2	1649.3	1696.5	1790.6	1885.0	1979.1	2073.4	2167.7	2261.9	2356.2	2450.3	2544.7
190	1691.2	1741.0	1790 8	1890.1	1989.7	2089.1	2188.6	2288.2	2387.5	2487.1	2586.5	2686.0
200	1780.2	1832.6	1885.0	1989.6	2094.4	2199.0	2303.8	2408.6	2513.2	2618.0	2722.6	2827.4
210	1869 2	1924.2	1979.3	2089.1	2199.1	2309.0	2419.0	2529.0	2638.9	2748.9	2858.7	2968.8
220	1958.2	2015.9	2073.5	2188.6	2303 8	2418.9	2534.2	2649.5	2764.5	2879.8	2994.9	3110.1
230	2047.2	2107.5	2167 8	2288.0	2408.6	2528.9	2649.4	2769.9	2890.2	3010.7	3131.0	3251.5
240	2136 2	2199 1	2262.0	2387.5	2513.3	2638.8	2764.6	2890.3	3015.8	3141.6	3267.1	3392.9
250	2225 3	2290 8	2356.3	2487.0	2618.0	2748.8	2879.8	3010.8	3141.6	3272.5	3403.3	3534.3
260	2314.3	2382.4	2450.5	2586.5	2722.7	2858.7	2994.9	3031.2	3267.2	3403.4	3539.4	3675.6
270	2403.3	2474.0	2544.8	2686.0	2827.4	2968.7	3110.1	3251 6	3392.8	3534.3	3675.5	3817.0
280	2492.3	2565 6	2639.0	2785.4	2932.2	3078.6	3225.3	3372.0	3518.5	3665.2	3811.6	3958.4
290	2581.3	2657.3	2733 3	2884.9	3036.9	3188.6	3340.5	3492.5	3644.1	3796.1	3947.8	4099.7
300	2670.3	2748.9	2827.5	2984.4	3141.6	3298.5	3455.7	3612.9	3769.8	3927.0	4083.9	4241.1
320	2848.3	2932.2	3016 0	3183.4	3351.0	3518.4	3686.1	3853.8	4021.1	4188.8	4356.2	4523.8
340	3026.3	3115.4	3204.5	3382.3	3560.5	3738.3	3916.5	4094.6	4272.4	4450.6	4628.4	4806.6
360	3204.4	3298 7	3393.0	3581.3	3769.9	3958.2	4146.8	4335.5	4523.8	4712.4	4900.7	5089.3
380	3382 4	3481.9	3581.5	3780.2	3979.4	4178.1	4377.2	4576.3	4775.1	4974.2	5172.9	5372.1
400	3560.0	3665.2	3770.0	3979.2	4188.8	4398.0	4607.6	4817.2	5026.4	5236.0	5445.2	5654.8
425	3782.9	3894.3	4005.6	4227.9	4450.6	4672.9	4895.5	5118.3	5340.5	5663.2	5785.5	—
450	4005.4	4123.4	4241.2	4476.6	4712.4	4947.8	5183.5	5419.4	5654.7	—	—	—
475	4227 9	4352.4	4476.8	4725.3	4974.2	5222.7	5471.5	5720.5	—	—	—	—
500	4450.5	4581.5	4712.5	4974.0	5236.0	5497.5	5759.5	—	—	—	—	—
525	4673.0	4810.6	4948.1	5222.7	5497.8	5772.4	—	—	—	—	—	—
550	4895.5	5039.7	5183.7	5471.4	5759.6	—	—	—	—	—	—	—
575	5118.0	5268.7	5419.3	5720.1	—	—	—	—	—	—	—	—
600	5340.6	5497.8	5655.0	—	—	—	—	—	—	—	—	—
650	5785.6	—	—	—	—	—	—	—	—	—	—	—
700	—	—	—	—	—	—	—	—	—	—	—	—
750	—	—	—	—	—	—	—	—	—	—	—	—
800	—	—	—	—	—	—	—	—	—	—	—	—

Table III—*Continued*

Velocity of Belt, Feet per Minute, for Different Pulley Diameters and Revolutions per Minute

R.P.M.	Diameter of Pulley, In.											
	56	58	60	66	72	78	84	90	96	102	108	120
10	—	—	—	—	—	—	—	—	251.3	267.0	282.7	314.2
20	293.2	303.7	314.2	345.6	377.0	408.4	439.8	471.2	502.7	534.1	565.5	628.3
30	439.8	455.5	471.2	518.4	565.5	612.6	659.7	706.9	754.0	801.1	848.2	942.5
40	586.4	607.4	628.3	691.2	754.0	816.8	879.6	942.5	1005.3	1068.1	1131.0	1256.6
50	733.1	759.2	785.4	864.0	942.5	1021.0	1099.6	1178.1	1256.7	1335.2	1413.7	1570.8
60	879.7	911.0	942.5	1036.7	1130.9	1225.2	1319.5	1413.7	1508.0	1602.2	1696.4	1885.0
70	1026.3	1062.9	1099.6	1209.5	1319.4	1429.4	1539.4	1649.3	1759.3	1869.2	1979.2	2199.1
80	1172.9	1214.7	1256.6	1382.3	1507.9	1633.6	1759.3	1885.0	2010.6	2136.2	2261.9	2513.3
90	1319.5	1366.6	1413.7	1555.1	1696.4	1837.8	1979.2	2120.6	2262.0	2403.3	2544.7	2827.4
100	1466.1	1518.4	1570.8	1727.9	1884.9	2042.0	2199.1	2356.2	2513.3	2670.3	2827.4	3141.6
110	1612.7	1670.2	1727.9	1900.7	2073.4	2246.2	2419.0	2591.8	2764.6	2937.3	3110.1	3455.8
120	1759.3	1822.1	1885.0	2073.5	2261.9	2450.4	2638.9	2827.4	3016.0	3204.4	3392.9	3769.9
130	1905.9	1973.9	2042.0	2246.3	2450.4	2654.6	2858.8	3063.1	3267.3	3471.4	3675.6	4084.1
140	2052.5	2125.8	2199.1	2419.1	2638.9	2858.8	3078.7	3298.7	3518.6	3738.4	3958.4	4398.2
150	2199.2	2277.6	2356.2	2591.9	2827.4	3063.0	3298.7	3534.3	3770.0	4005.5	4241.1	4712.4
160	2345.8	2429.4	2513.3	2764.6	3015.8	3267.2	3518.6	3769.9	4021.3	4272.5	4523.8	5026.6
170	2492.4	2581.3	2670.4	2937.4	3204.3	3471.4	3738.5	4005.5	4272.6	4539.5	4806.6	5340.7
180	2639.0	2733.1	2827.4	3110.2	3392.8	3675.6	3958.4	4241.2	4523.9	4806.5	5089.3	5654.9
190	2785.6	2885.0	2984.5	3283.0	3581.3	3879.8	4178.3	4476.8	4775.3	5073.6	5372.1	—
200	2932.2	3036.8	3141.6	3455.8	3769.8	4084.0	4398.2	4712.4	5026.6	5340.6	5654.8	—
210	3078.8	3188.6	3298.7	3628.6	3958.3	4288.2	4618.1	4948.0	5277.9	5607.6	—	—
220	3225.4	3340.5	3455.8	3801.4	4146.8	4492.4	4838.0	5183.6	5529.3	—	—	—
230	3372.0	3492.3	3612.8	3974.2	4335.3	4696.0	5057.9	5419.3	5780.6	—	—	—
240	3518.6	3644.2	3769.9	4147.0	4523.8	4900.8	5277.8	5654.9	—	—	—	—
250	3665.3	3796.0	3927.0	4319.8	4712.3	5105.0	5497.8	5890.5	—	—	—	—
260	3811.9	3947.8	4084.1	4492.5	4900.7	5309.2	5717.7	—	—	—	—	—
270	3958.5	4099.7	4241.2	4665.3	5089.2	5513.4	5937.6	—	—	—	—	—
280	4105.1	4251.5	4398.2	4838.1	5277.7	5717.6	—	—	—	—	—	—
290	4241.7	4403.4	4555.3	5010.9	5466.2	—	—	—	—	—	—	—
300	4398.3	4555.2	4712.4	5183.7	5654.7	—	—	—	—	—	—	—
320	4691.5	4858.9	5026.6	5529.3	—	—	—	—	—	—	—	—
340	4984.7	5162.6	5340.7	—	—	—	—	—	—	—	—	—
360	5278.0	5466.2	5654.9	—	—	—	—	—	—	—	—	—
380	5571.2	5769.9	—	—	—	—	—	—	—	—	—	—
400	—	—	—	—	—	—	—	—	—	—	—	—

Table IV. —Cross-Sectional Area of Belts, Square Inches

Width of Belt, In.	Thickness of Belt, In.															
	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
1	0.125	0.188	0.25	0.313	0.375	0.438	0.50	0.563	0.625	0.688	0.75	0.813	0.875	0.938	1.00
1 1/2	0.188	0.281	0.375	0.469	0.563	0.656	0.75	0.844	0.938	1.031	1.125	1.219	1.313	1.406	1.50
2	0.25	0.375	0.50	0.625	0.75	0.875	1.00	1.125	1.25	1.375	1.50	1.625	1.75	1.875	2.00
2 1/2	0.313	0.469	0.625	0.779	0.938	1.094	1.25	1.407	1.563	1.719	1.875	2.031	2.188	2.344	2.50
3	0.375	0.563	0.75	0.938	1.125	1.313	1.50	1.688	1.875	2.063	2.25	2.438	2.625	2.813	3.00
3 1/2	0.438	0.656	0.875	1.094	1.313	1.521	1.75	1.969	1.188	2.406	2.625	2.844	3.063	3.281	3.50
4	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00	3.25	3.50	3.75	4.00
4 1/2	0.563	0.844	1.125	1.404	1.688	1.969	2.25	2.532	2.813	3.094	3.375	3.656	3.938	4.219	4.50
5	0.625	0.938	1.25	1.563	1.875	2.188	2.50	2.813	3.125	3.438	3.75	4.063	4.375	4.688	5.00
5 1/2	0.688	1.032	1.375	1.717	2.063	2.407	2.75	3.095	3.438	3.782	4.125	4.469	4.813	5.157	5.50
6	0.75	1.125	1.50	1.875	2.25	2.625	3.00	3.375	3.75	4.125	4.50	4.875	5.25	5.625	6.00
7	0.875	1.313	1.75	2.188	2.625	3.063	3.50	3.938	4.375	4.813	5.25	5.688	6.125	6.563	7.00
8	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00	5.50	6.00	6.50	7.00	7.50	8.00
9	1.125	1.688	2.25	2.813	3.375	3.938	4.50	5.063	5.625	6.188	6.75	7.313	7.875	8.438	9.00
10	1.25	1.875	2.50	3.125	3.75	4.375	5.00	5.625	6.25	6.875	7.50	8.125	8.75	9.375	10.00
11	1.375	2.063	2.75	3.438	4.125	4.813	5.50	6.188	6.875	7.563	8.25	8.938	9.625	10.313	11.00
12	1.50	2.25	3.00	3.75	4.50	5.25	6.00	6.75	7.50	8.25	9.00	9.75	10.50	11.25	12.00
14	1.75	2.625	3.50	4.375	5.25	6.125	7.00	7.875	8.75	9.625	10.50	11.375	12.25	13.125	14.00
16	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00	12.00	13.00	14.00	15.00	16.00
18	2.25	3.375	4.50	5.625	6.75	7.875	9.00	10.125	11.25	12.375	13.50	14.625	15.75	16.875	18.00
20	2.50	3.75	5.00	6.25	7.50	8.75	10.00	11.25	12.50	13.75	15.00	16.25	17.50	18.75	20.00
22	2.75	4.125	5.50	6.875	8.25	9.625	11.00	12.375	13.75	15.125	16.50	17.875	19.25	20.625	22.00
24	3.00	4.50	6.00	7.50	9.00	10.50	12.00	13.50	15.00	16.50	18.00	19.50	21.00	22.50	24.00
26	3.25	4.875	6.50	8.125	9.75	11.375	13.00	14.625	16.25	17.875	19.50	21.125	22.75	24.375	26.00
28	3.50	5.250	7.00	8.75	10.50	12.25	14.00	15.75	17.50	19.25	21.00	22.75	24.50	26.25	28.00
30	3.75	5.625	7.50	9.375	11.25	13.125	15.00	16.875	18.75	20.625	22.50	24.375	26.25	28.125	30.00
36	4.50	6.75	9.00	11.25	13.50	15.75	18.00	20.25	22.50	24.75	27.00	29.25	31.50	33.75	36.00
42	5.25	7.875	10.50	13.125	15.75	18.375	21.00	23.625	26.25	28.875	31.50	34.125	36.75	39.375	42.00
48	6.00	9.00	12.00	15.00	18.00	21.00	24.00	27.00	30.00	33.00	36.00	39.00	42.00	45.00	48.00
54	6.75	10.125	13.50	16.875	20.25	23.625	27.00	30.375	33.75	37.125	40.50	43.875	47.25	50.625	54.00
60	7.50	11.25	15.00	18.75	22.50	26.25	30.00	33.75	37.50	41.25	45.00	48.75	52.50	56.25	60.00
66	8.25	12.375	16.50	20.625	24.75	28.875	33.00	37.125	41.25	45.375	49.50	53.625	57.75	61.875	66.00
72	9.00	13.50	18.00	22.50	27.00	31.50	36.00	40.50	45.00	49.50	54.00	58.50	63.00	67.50	72.00

Table V

Diameter, Circumference, and Rim Velocity of Pulleys

Diameter, In.	Circumference, In.	Circumference, Ft.	Diameter, In.	Circumference, In.	Circumference, Ft.	Diameter, In.	Circumference, In.	Circumference, Ft.	Diameter, In.	Circumference, In.	Circumference, Ft.
1	3.142	0.262	25	78.540	6.545	49	153.938	12.828	73	229.336	19.111
1½	4.712	.393	25½	80.111	6.676	49½	155.509	12.959	73½	230.907	19.242
2	6.283	.524	26	81.681	6.807	50	157.080	13.090	74	232.478	19.373
2½	7.854	.655	26½	83.252	6.938	50½	158.650	13.221	74½	234.049	19.504
3	9.425	.785	27	84.823	7.069	51	160.221	13.352	75	235.619	19.635
3½	10.996	.916	27½	86.394	7.199	51½	161.792	13.483	75½	237.190	19.766
4	12.566	1.047	28	87.965	7.330	52	163.363	13.614	76	238.761	19.897
4½	14.137	1.178	28½	89.535	7.461	52½	164.934	13.745	76½	240.332	20.028
5	15.708	1.309	29	91.106	7.592	53	166.504	13.875	77	241.903	20.159
5½	17.279	1.440	29½	92.677	7.723	53½	168.075	14.006	77½	243.473	20.289
6	18.850	1.571	30	94.248	7.854	54	169.646	14.137	78	245.044	20.420
6½	20.420	1.702	30½	95.819	7.985	54½	171.217	14.268	78½	246.615	20.551
7	21.991	1.833	31	97.389	8.116	55	172.788	14.399	79	248.186	20.682
7½	23.562	1.964	31½	98.960	8.247	55½	174.358	14.530	79½	249.757	20.813
8	25.133	2.094	32	100.531	8.378	56	175.929	14.661	80	251.327	20.944
8½	26.704	2.225	32½	102.102	8.509	56½	177.500	14.792	80½	252.898	21.075
9	28.274	2.356	33	103.673	8.639	57	179.071	14.923	81	254.469	21.206
9½	29.845	2.487	33½	105.243	8.770	57½	180.642	15.054	81½	256.040	21.337
10	31.416	2.618	34	106.814	8.901	58	182.212	15.184	82	257.611	21.468
10½	32.987	2.749	34½	108.385	9.032	58½	183.783	15.315	82½	259.181	21.598
11	34.558	2.880	35	109.956	9.163	59	185.354	15.446	83	260.752	21.729
11½	36.128	3.011	35½	111.527	9.294	59½	186.925	15.577	83½	262.323	21.860
12	37.699	3.142	36	113.097	9.425	60	188.496	15.708	84	263.894	21.991
12½	39.270	3.273	36½	114.668	9.556	60½	190.066	15.839	85	267.035	22.253
13	40.841	3.403	37	116.239	9.687	61	191.637	15.969	86	270.177	22.516
13½	42.412	3.534	37½	117.810	9.818	61½	193.208	16.101	87	273.319	22.777
14	43.982	3.665	38	119.381	9.949	62	194.779	16.232	88	276.460	23.038
14½	45.553	3.796	38½	120.951	10.079	62½	196.350	16.363	89	279.602	23.300
15	47.124	3.927	39	122.522	10.210	63	197.920	16.493	90	282.743	23.562
15½	48.695	4.058	39½	124.093	10.341	63½	199.491	16.624	91	285.885	23.824
16	50.265	4.189	40	125.664	10.472	64	201.062	16.755	92	289.027	24.086
16½	51.836	4.320	40½	127.235	10.603	64½	202.633	16.886	93	292.168	24.347
17	53.407	4.451	41	128.805	10.734	65	204.204	17.017	94	295.310	24.609
17½	54.978	4.582	41½	130.376	10.865	65½	205.774	17.148	95	298.451	24.871
18	56.549	4.712	42	131.947	10.996	66	207.345	17.279	96	301.593	25.133
18½	58.119	4.843	42½	133.518	11.127	66½	208.916	17.410	97	304.734	25.395
19	59.690	4.974	43	135.088	11.257	67	210.487	17.541	98	307.876	25.656
19½	61.261	5.105	43½	136.659	11.388	67½	212.058	17.672	99	311.018	25.918
20	62.832	5.236	44	138.230	11.519	68	213.628	17.802	100	314.159	26.180
20½	64.403	5.367	44½	139.801	11.650	68½	215.199	17.933	101	317.302	26.442
21	65.973	5.498	45	141.372	11.781	69	216.770	18.064	102	320.443	26.704
21½	67.544	5.629	45½	142.942	11.912	69½	218.341	18.195	103	323.585	26.965
22	69.115	5.759	46	144.513	12.043	70	219.911	18.326	104	326.726	27.227
22½	70.686	5.891	46½	146.084	12.174	70½	221.482	18.457	105	329.868	27.489
23	72.257	6.021	47	147.655	12.305	71	223.053	18.588	106	333.010	27.751
23½	73.827	6.152	47½	149.226	12.436	71½	224.624	18.719	107	336.151	28.013
24	75.398	6.283	48	150.796	12.566	72	226.195	18.850	108	339.293	28.274
24½	76.969	6.414	48½	152.367	12.697	72½	227.765	18.980			

To obtain rim velocity in feet per minute, multiply the circumference in feet by the number of revolutions per minute.

Table VI

Width of Pulley to be Used with a Given Width of Belt

(Table calculated from Barth's formulæ. See page 73.)

Width of Belt, In.	Width of Pulley, In.		Width of Belt, In.	Width of Pulley, In.		Width of Belt, In.	Width of Pulley, In.	
	Formula, $P = 1\frac{3}{32}B + \frac{3}{16}$.	Formula,* $P = 1\frac{3}{16}B + \frac{3}{8}$.		Formula, $P = 1\frac{3}{32}B + \frac{3}{16}$.	Formula,* $P = 1\frac{3}{16}B + \frac{3}{8}$.		Formula, $P = 1\frac{3}{32}B + \frac{3}{16}$.	Formula,* $P = 1\frac{3}{16}B + \frac{3}{8}$.
1	1 9/32	1 9/16	7	7 27/32	8 11/16	24	26 7/16	28 7/8
1 1/2	1 53/64	2 5/32	8	8 15/16	9 7/8	26	28 5/8	31 1/4
2	2 3/8	2 3/4	9	10 1/32	11 1/16	28	30 13/16	33 5/8
2 1/2	2 59/64	3 11/32	10	11 1/8	12 1/4	30	33	36
3	3 15/32	3 15/16	11	12 7/32	13 7/16	36	39 9/16	43 1/8
3 1/2	4 1/64	4 17/32	12	13 5/16	14 5/8	42	46 1/8	50 1/4
4	4 9/16	5 1/8	14	15 1/2	17	48	52 11/16	57 3/8
4 1/2	5 7/64	5 23/32	16	17 11/16	19 3/8	54	59 1/4	64 1/2
5	5 21/32	6 5/16	18	19 7/8	21 3/4	60	65 13/16	71 5/8
5 1/2	6 13/64	6 29/32	20	22 1/16	24 1/8	66	72 3/8	78 3/4
6	6 3/4	7 1/2	22	24 1/4	26 1/2	72	78 15/16	85 7/8

* Preferred formula.

Table V

Diameter, Circumference, and Rim Velocity of Pulleys

Diameter, In.	Circumference, In.	Circumference, Ft.	Diameter, In.	Circumference, In.	Circumference, Ft.	Diameter, In.	Circumference, In.	Circumference, Ft.	Diameter, In.	Circumference, In.	Circumference, Ft.
1	3.142	0.262	25	78.540	6.545	49	153.938	12.828	73	229.336	19.111
1 1/2	4.712	.393	25 1/2	80.111	6.676	49 1/2	155.509	12.959	73 1/2	230.907	19.242
2	6.283	.524	26	81.681	6.807	50	157.080	13.090	74	232.478	19.373
2 1/2	7.854	.655	26 1/2	83.252	6.938	50 1/2	158.650	13.221	74 1/2	234.049	19.504
3	9.425	.785	27	84.823	7.069	51	160.221	13.352	75	235.619	19.635
3 1/2	10.996	.916	27 1/2	86.394	7.199	51 1/2	161.792	13.483	75 1/2	237.190	19.766
4	12.566	1.047	28	87.965	7.330	52	163.363	13.614	76	238.761	19.897
4 1/2	14.137	1.178	28 1/2	89.535	7.461	52 1/2	164.934	13.745	76 1/2	240.332	20.028
5	15.708	1.309	29	91.106	7.592	53	166.504	13.875	77	241.903	20.159
5 1/2	17.279	1.440	29 1/2	92.677	7.723	53 1/2	168.075	14.006	77 1/2	243.473	20.289
6	18.850	1.571	30	94.248	7.854	54	169.646	14.137	78	245.044	20.420
6 1/2	20.420	1.702	30 1/2	95.819	7.985	54 1/2	171.217	14.268	78 1/2	246.615	20.551
7	21.991	1.833	31	97.389	8.116	55	172.788	14.399	79	248.186	20.682
7 1/2	23.562	1.964	31 1/2	98.960	8.247	55 1/2	174.358	14.530	79 1/2	249.757	20.813
8	25.133	2.094	32	100.531	8.378	56	175.929	14.661	80	251.327	20.944
8 1/2	26.704	2.225	32 1/2	102.102	8.509	56 1/2	177.500	14.792	80 1/2	252.898	21.075
9	28.274	2.356	33	103.673	8.639	57	179.071	14.923	81	254.469	21.206
9 1/2	29.845	2.487	33 1/2	105.243	8.770	57 1/2	180.642	15.054	81 1/2	256.040	21.337
10	31.416	2.618	34	106.814	8.901	58	182.212	15.184	82	257.611	21.468
10 1/2	32.987	2.749	34 1/2	108.385	9.032	58 1/2	183.783	15.315	82 1/2	259.181	21.598
11	34.558	2.880	35	109.956	9.163	59	185.354	15.446	83	260.752	21.729
11 1/2	36.128	3.011	35 1/2	111.527	9.294	59 1/2	186.925	15.577	83 1/2	262.323	21.860
12	37.699	3.142	36	113.097	9.425	60	188.496	15.708	84	263.894	21.991
12 1/2	39.270	3.273	36 1/2	114.668	9.556	60 1/2	190.066	15.839	85	267.035	22.253
13	40.841	3.403	37	116.239	9.687	61	191.637	15.969	86	270.177	22.516
13 1/2	42.412	3.534	37 1/2	117.810	9.818	61 1/2	193.208	16.101	87	273.319	22.777
14	43.982	3.665	38	119.381	9.949	62	194.779	16.232	88	276.460	23.038
14 1/2	45.553	3.796	38 1/2	120.951	10.079	62 1/2	196.350	16.363	89	279.602	23.300
15	47.124	3.927	39	122.522	10.210	63	197.920	16.493	90	282.743	23.562
15 1/2	48.695	4.058	39 1/2	124.093	10.341	63 1/2	199.491	16.624	91	285.885	23.824
16	50.265	4.189	40	125.664	10.472	64	201.062	16.755	92	289.027	24.086
16 1/2	51.836	4.320	40 1/2	127.235	10.603	64 1/2	202.633	16.886	93	292.168	24.347
17	53.407	4.451	41	128.805	10.734	65	204.204	17.017	94	295.310	24.609
17 1/2	54.978	4.582	41 1/2	130.376	10.865	65 1/2	205.774	17.148	95	298.451	24.871
18	56.549	4.712	42	131.947	10.996	66	207.345	17.279	96	301.593	25.133
18 1/2	58.119	4.843	42 1/2	133.518	11.127	66 1/2	208.916	17.410	97	304.734	25.395
19	59.690	4.974	43	135.088	11.257	67	210.487	17.541	98	307.876	25.656
19 1/2	61.261	5.105	43 1/2	136.659	11.388	67 1/2	212.058	17.672	99	311.018	25.918
20	62.832	5.236	44	138.230	11.519	68	213.628	17.802	100	314.159	26.180
20 1/2	64.403	5.367	44 1/2	139.801	11.650	68 1/2	215.199	17.933	101	317.302	26.442
21	65.973	5.498	45	141.372	11.781	69	216.770	18.064	102	320.443	26.704
21 1/2	67.544	5.629	45 1/2	142.942	11.912	69 1/2	218.341	18.195	103	323.585	26.965
22	69.115	5.759	46	144.513	12.043	70	219.911	18.326	104	326.726	27.227
22 1/2	70.686	5.891	46 1/2	146.084	12.174	70 1/2	221.482	18.457	105	329.868	27.489
23	72.257	6.021	47	147.655	12.305	71	223.053	18.588	106	333.010	27.751
23 1/2	73.827	6.152	47 1/2	149.226	12.436	71 1/2	224.624	18.719	107	336.151	28.013
24	75.398	6.283	48	150.796	12.566	72	226.195	18.850	108	339.293	28.274
24 1/2	76.969	6.414	48 1/2	152.367	12.697	72 1/2	227.765	18.980			

To obtain rim velocity in feet per minute, multiply the circumference in feet by the number of revolutions per minute.